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FACILITY FORM 802

14-000000-22222
(ACCESSION NUMBER)
301
(PAGES)
CR-83516
(NASA CR OR TMX OR AD NUMBER)

(THRU)
1
(CODE)
04
(CATEGORY)



PHILCO-FORD CORPORATION
Aeronutronic Division
Newport Beach, California

FINAL REPORT

AUTOMATED BIOLOGICAL LABORATORY
SOIL SAMPLING STUDY

Prepared for: Bioscience Programs Division
 Office of Space Science and Applications
 NASA Headquarters
 Washington, D. C.

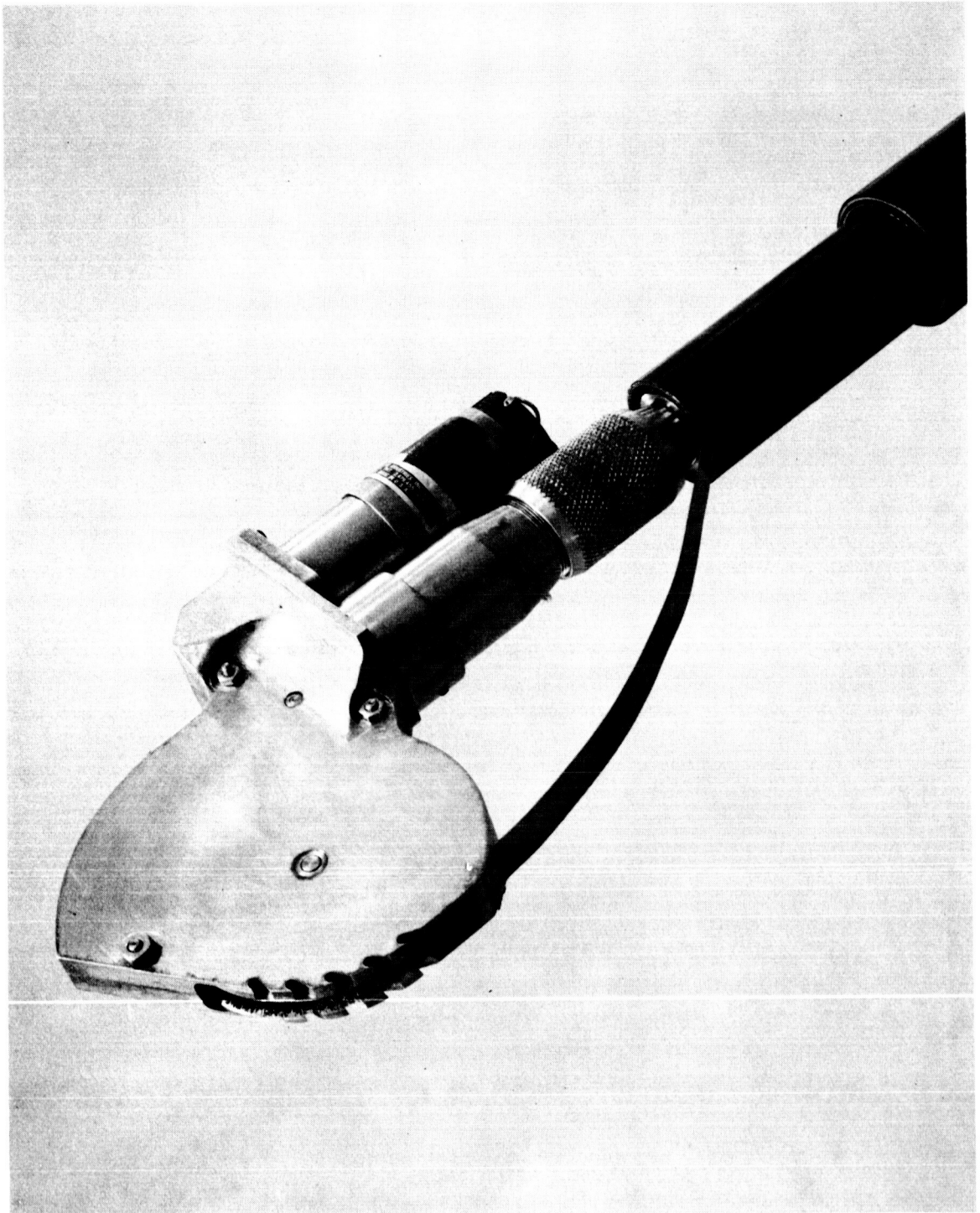
Under Contract: NASw-1065

Reporting Period: 17 February 1966 - 17 February 1967

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17 February 1967





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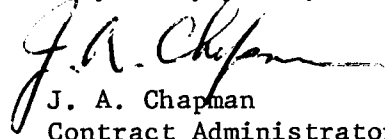
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ABSTRACT

This report describes the conduct and results of a three phase program proceeding from the conceptual design of a number of sample handling systems to the fabrication and testing of two selected engineering prototype systems. The sample handling systems are capable of acquiring, transporting, processing and transferring small particulate samples to the analytical instruments to be employed within Voyager-class biological capsule configurations for delivery to the surface of Mars in the 1970 decade. The concepts and prototype systems consider two basic configurations: (1) samplers capable of some appreciable deployment distance from a capsule (horizontally deployed sample acquisition systems) and (2) samplers deployed radially from a bio-capsule to the surface and subsurface (vertically deployed systems).

The program proceeded in three phases: (1) conceptual design study phase, (2) breadboard test fixture, and (3) engineering prototype development phase. The horizontally deployed sample handling system developed consists of a rotating wire brush deployed on a telescoping boom containing a combined pneumatic and mechanical sample transporting and collection mode. Sample sizing and processing is accomplished by the wire brush during the acquisition of a soil sample. The transfer of the sample to the analytical instruments can be accomplished by gravity drop stages and/or by omnidirectional pneumatic tubes or helical screw conveyors. The vertically deployed engineering prototype system is a rotating conical sieve which is dumped by rapid spinning and vibration caused by an eccentric weight. The sample is sized both during acquisition and by means of a tilted sieve within the dump chamber. Transfer of the processed sample is accomplished as described above. Tests have proved that both samplers are capable of acquiring and transporting particulate samples individually weighing one gram or more from a variety of soil models ranging from firmly cemented hardpan through loose rubble and sand to compacted silt.

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LIST OF SYMBOLS

C_D	- Drag coefficient
D	- Drum diameter, inches
E	- Young's Modulus of Elasticity, lbs/in. ²
I	- Cross-section moment of inertia, in. ⁴
M	- Bending moment, in.-lbs
M_{cr}	- Critical buckling moment, in.-lbs
N	- Motor or compressor speed, rpm
P	- Concentrated loading, lbs
R	- Universal gas constant
S	- Effective drag area, ft ²
T	- Torsional moment, in.-lbs
V	- Volume, in. ³
W	- Weight, lbs
d	- Tube diameter, inches
k	- Ratio of specific heats
l	- Boom length, inches
p	- pressure, lbs/in. ²
q	- Dynamic pressure, lbs/ft ²
r	- Tube radius, inches
t	- Wall thickness, inches
v	- Wind velocity, ft/sec
\dot{w}	- Flow rate, lbs/min
w	- Distributed loading, lbs/in.
δ	- Vertical deflection, inches
θ	- Torsional deflection, radians
μ	- Poisson's ratio
ρ	- Material density, lbs/in. ³
σ_c	- Compressive yield strength, lbs/in. ⁻²
σ	- Axial stress, lbs/in. ²
τ	- Shear stress, lbs/in. ²

ACKNOWLEDGEMENTS

This ABL follow-on program brought together several disciplines for the development of concepts for biological material samplers and the design, fabrication and testing of engineering prototypes of these devices. The initial contributions of the JPL bioscience group under Dr. Gerald Soffen and the geoscience people at JPL, represented by Dr. Douglas Nash, were important in defining the problems besetting biological material samplers and the post - Mariner IV surface characteristics of Mars. Messrs. George Hotz and Earl Howard, the JPL technical monitors, provided constructive direction and exchanged sampling experience to aid the conduct of the program.

The program was initiated by the Aeronutronic Division of Philco Corporation under the direction of the Space and Re-entry Systems Operation. Mr. William Hostetler was Program Manager with Mr. Temple Neumann as Program Engineer. A change in corporate structure occurred in September 1966, so the program was assigned to the Lunar and Planetary Programs of the Space and Re-entry Systems Division of Philco-Ford Corporation with Mr. Robert S. Kraemer, Manager, and Mr. Wilfred H. Bachle, Program Engineer. The principal contributors to the program include:

Mr. Wilfred H. Bachle	Supervisor, Advanced Design Space Physics Department Applied Research Laboratory
Mr. A. O. Brodie	Design and Testing
Mr. R. A. Smith	Testing and Model Construction
Mr. R. R. Woodcock	Design
Mr. G. P. Zebal	Model Definition and Testing

SECTION 1

INTRODUCTION

This report contains the results of a conceptual design study of sample handling systems and the development of two prototype systems capable of acquiring and then transporting small particulate samples to instruments employed within early Voyager-class biological capsules. The work was performed by Space and Re-Entry Systems Division, Newport Beach Operations, Philco-Ford Corporation, for the Bioscience Program Division, the National Aeronautics and Space Administration, over a period of eleven months under NASA Contract NASW-1065, Supplemental Agreement No. 2. Technical direction was provided by the Space Sciences Division, Jet Propulsion Laboratory.

1.1 PROGRAM OBJECTIVES AND CONDUCT

Program objectives were basically twofold: to conduct conceptual design studies of a number of sampling systems from which one or more types of sample collecting and transporting devices would be selected and to design, breadboard, develop and test the selected prototype sample handling systems. By definition, a sample handling system shall include sample acquisition, transport, processing and transfer devices. System concepts were to consider two basic types of sample acquiring devices:

- (1) Vertically deployed samplers, operating radially around the circumference of the biocapsule, able to sample both the surface and shallow subsurface.
- (2) Horizontally deployed samplers, capable of operating an appreciable distance from the biocapsule, that should be designed to a high degree of sample acquisition and transport reliability.

The program was conducted in three phases:

- (1) Conceptual design study,
- (2) Breadboard test fixture fabrication and testing,
and
- (3) Engineering prototype design, development and testing.

The conceptual design study phase was initiated by an evaluation of previous efforts by others to design and develop general purpose sample acquiring systems. A literature survey, a seminar attended by Aeronutronic technical specialists, and an analysis of JPL in-house and sponsored studies and concepts was conducted. A bibliography is presented in Appendix A. Thirty-two (32) preliminary concepts, comprising Appendix B of this report, were presented to JPL representatives at the Preliminary Design Review Meeting of 6 April 1966. These representatives evaluated the concepts and designated the eight most promising for preliminary design treatment. The eight preliminary design layouts are included in Appendix C. By letter of 20 June 1966, JPL and Aeronutronic mutually agreed that the following two preliminary designs should be pursued into breadboard testing and detailed engineering design phases:

- (1) Vertically deployed conical abrading sieve and spin dump concept.
- (2) Horizontally deployed rotating wire brush on boom concept.

During this conceptual phase, engineering studies and parametric analyses were also conducted on deployable boom configurations and materials, supersonic jets, pneumatic motors, and gas flow characteristics within pneumatic transport tubes at ambient and low atmospheric pressures.

The breadboard test fixture phase was actually commenced prior to the conclusion of the conceptual design phase. A laboratory test setup was made for the rotating brush concept to obtain some qualitative performance characteristics. A three segment boom utilizing telescoping aluminum tubing was fabricated in order to examine extension rates and bending deflection values when pneumatically extending the boom. The four soil models stipulated by JPL were collected or manufactured; bins for testing the conical abrasive sieve were obtained and filled. Breadboard tests of the rotating wire brush test fixture in rotating wooden bins were conducted to examine floating and fixed head configurations, brush geometry and power and weight requirements, sample pickup and holding characteristics, and overall system geometry versus soil surface interactions. Two pneumatic transport tubes were fabricated from transparent acrylic

material to allow visual observation of particles being transported. One tube diverged to match the internal dimensions of a telescoping boom; the other was a constant diameter acrylic tube. In both tubes flow was produced by a single vaneaxial blower attached to the exit of a cyclone collector or by the addition of another vaneaxial blower to the entrance of the tube. Tests were conducted at room pressure and at reduced pressures (to 5 mb) within the NASA-owned Mars simulation chamber. During this phase final designs of the rotating brush and spin dumped conical sieve were initiated and essentially completed.

The final engineering prototype development phase saw the completion of final design and top assembly drawings, the fabrication of the prototype samplers and the deployment and sampler transporting accessories, and their testing on both a subsystem and complete system basis.

1.2 SAMPLE HANDLING SYSTEM REQUIREMENTS AND CONSTRAINTS

The requirements for the sample to be delivered to the biocapsule and the sample handling system design goals are outlined in Table 1.1, as follows:

TABLE 1.1

SAMPLE REQUIREMENTS AND SAMPLE HANDLING SYSTEM DESIGN GOALS

<u>Sample Requirements</u>	<u>System Design Goals</u>
1. Several grams per sample.	1. Weight - approximately 3 lbs.
2. Several samples from several different locations, if feasible.	2. Power - 25 watts for several minutes maximum.
3. Maximum particle size - 1000 μ .	3. Volume - 25 in. ³ maximum
4. Minimum particle size - none.	4. Acquisition of surface skin and subsurface materials, other than solid rock, to a depth of several millimeters.
5. Elimination of heat degradation.	
6. Elimination of contamination.	
7. Elimination of anti-biological sorting.	

In addition to these specific requirements and design goals, the following factors were considered:

- (1) Hardening to impact levels of 2000 to 10,000 g.

- (2) Meeting JPL sterilization requirements in accordance with Voyager specifications.
- (3) Operating within a temperature range of -100°F to 150°F .
- (4) Penetrating a considerable thickness of impact absorbing material prior to initial deployment.
- (5) Acquiring a sample from a non-gravity oriented platform.

Finally, deployment mechanisms should be designed for biocapsules having a maximum radius (spherical) of 3 feet. The handling systems should consider an atmospheric pressure of 5 to 10 mb. For design purposes and testing media, the following surface material models should be utilized:

- (1) Cohesive powder
- (2) Cohesionless particulate material (sand)
- (3) Hardpan
- (4) Loose rubble

1.3 SUMMARY OF RESULTS AND TECHNICAL CONCLUSIONS

This program has resulted in the development of two selected general purpose prototype sample handling systems. The prototype samplers conform to the system requirements and constraints stipulated with NASA Contract NASW-1065, Supplemental Agreement No. 2. The preliminary mechanical design configuration drawings and the final mechanical configuration drawings are being furnished in a package separate from this report but included as if here appended.

The selected vertically and horizontally deployed prototype samples have been subjected to a parametric series of tests sufficient to demonstrate their compliance to the performance and reliability goals stipulated by the contract. The prototype test matrices are included in detail in Section 4.1. The test program served to debug the prototype samplers and to establish basic performance criteria in relation to the stipulated models.

The results of the testing phase of this program lead to the recommendation that further testing of these prototype sample handling systems is warranted. These proposed additional tests can utilize both presently available

prototype equipment and test modified prototypes to further general purpose sampler technology and to define the specific performance of prototype systems in the following areas:

- (1) Establishing performance in the currently defined Martian environment.
- (2) Expanding the range of soil models in order to identify the range of materials that can be effectively collected.
- (3) Determining biological effectiveness of the prototype samplers.
- (4) Conducting non-operating tests to assess the ability of the sample handling systems to withstand sterilization and mission environments and to determine their working lifetime limit.
- (5) Finally, conducting field tests under extreme natural site conditions.

SECTION 2

CONCEPTUAL DESIGN PHASE

The purpose of the Conceptual Design Phase of the program was to generate a number of ideas, and therefrom conceptually design a series of general purpose sample handling systems capable of supplying small size particulate samples to the instruments within Voyager-class biological capsules. The phase was conducted in three distinct endeavors:

- (1) Literature survey and review of techniques developed by JPL and other workers.
- (2) Cooperative definition of test models, pre-ABL Voyager sampling rationale and evaluation criteria for rating sampler concepts.
- (3) Development of conceptual designs of sample handling systems.

By definition, a sample handling system should include sample acquisition device(s), sample transport mechanism, sample processor (if it becomes necessary to eliminate particles $> 1000\mu$ or $< 1\mu$), and a sample transfer device (for final delivery to an appropriate container within the bio-capsule). The sample requirements and operational constraints and considerations for a sample handling system have been described in paragraph 1.2 and Table 1.I.

Prior to the actual initiation of this phase during the period 11 September 1965 to 31 January 1966, several meetings were held with personnel from the Bioscience Programs Division, NASA Headquarters, and from the Space Sciences Division, Jet Propulsion Laboratory, to define the objectives of an Automated Biological Laboratory follow-on effort. It was finally agreed

that general purpose sample handling systems suitable for early Voyager-class Mars landing missions were of fundamental value to the ABL program. To this end a series of tasks covering the analysis, design, fabrication and testing of both vertically and horizontally deployed sampling systems that would conform to general size, weight, power and performance constraints were described and embodied in the ABL Study Contract, Supplemental Agreement No. 2, dated 17 February 1966.

2.1 PRELIMINARY STUDIES AND DATA ACQUISITION

A literature search was conducted by the Aeronutronic Technical Library staff. This bibliography and other technical treatises dealing with the engineering requirements for obtaining and transporting particulate samples within the Mars environment were reviewed by Philco-Ford design engineers. A selected bibliography is included as Appendix A to this report. In conjunction with the literature survey, the geological sampling techniques under development and concepts under study by the Soil Sampling Laboratory at JPL under the supervision of Mr. George M. Hotz were reviewed. Descriptions of pneumatic sampling devices and biologically oriented techniques developed under Dr. Gerald A. Soffen and Mr. Jerry L. Stuart of JPL were received by Philco-Ford personnel. Many of these techniques and modifications thereof are identified and incorporated in the list of conceptual designs submitted to JPL for review and evaluation and included within the appendices to this report.

Dr. Soffen and Dr. Douglas B. Nash of JPL defined six representative soil models and corresponding laboratory test materials that should span the general ranges of the uppermost surface materials expected to be encountered on Mars. These models were proposed for testing the sample handling systems to be developed by this program. Descriptions of the models, their physical characteristics, mode of formation on Mars, and recommended test material equivalents are given on Table 2.I. From this list, the following four surface models were selected and incorporated within the Supplemental Agreement:

- (1) Cohesive powder
- (2) Cohesionless particulate material (noncohesive sand)
- (3) Hardpan
- (4) Loose rubble

The specific requirements and design goals for sampling systems were outlined in Table 1.I. In addition to these specific items many factors effecting the design of sample handling systems were to be considered.

TABLE 2.I
SURFACE MODELS

<u>Surface Model</u>	<u>Approximate Physical Characteristics</u>	<u>Possible Geological Mode of Formation on Mars</u>	<u>Examples of Earth Materials for Test Purposes</u>
1. Cemented powder	Particles less than about 50 μ in size which are lightly to moderately cemented	a. Iron-oxide coated and cemented silt b. Sputter-cemented silt	Hardpan or adobe
2. Cohesive fine powder	Particles of size between 5-10 μ	Wind blown and deposited particles	Silt or loess
3. Noncohesive sand	Particles between 100-500 μ	Wind sorted particles	Sand; may include silicate-oxide, halide, carbonate, nitrate, and sulfate minerals
4. Rubble	Mixture of fragmented particles of all sizes less than about 10 cm	Impact-pulverized bedrock	Crushed unsorted igneous rock, such as basalt (Little Lake basalt)
5. Friable, porous rock	Lithified volcanic ash particles of all sizes less than about 4 mm	Viscous volcanic magma comminuted by effervescence	Rhyolite tuff (Bishop tuff)
6. Solid bedrock	Massive crystalline rock, fine to medium grain size. May or may not be slightly to moderately vesicular	a. Surface lava flow b. Exposed subsurface intrusive rock	Basalt (Little Lake)

NOTE: Subsurface layered models could consist of layers of any one, or various combinations, of 1-4 overlying 5 and/or 6. Layer thicknesses could vary from 1 μ to 1 km, or more.

These factors included hardening, sterilization, biocapsule radius, penetration of impact limiting material, and sampling from a sloping platform. Martian operating environment constraints were twofold:

- (1) Temperature range, minus 100°F to 150°F
- (2) Atmospheric pressure range, 5 to 10 mb

Although the fundamental objective of pre-ABL Voyager-class sampling systems is biological, the geological data obtained therefrom will be of great value; therefore, a general purpose sampler must consider both areas. To this end a rationale was developed by JPL to guide the conceptual design study beyond the letter of the specific requirements, design goals, constraints and considerations identified in the Statement of Work. The pre-ABL Voyager sampling rationale is outlined as follows:

- I. The basic objective of the pre-ABL Voyager sampling task is biological sampling; however, as it is recognized that there are many elements common both to biological and geological sampling, this task should be concerned with both. The only exception to the foregoing is that the case of sampling in solid bedrock, of little or no interest biologically, but of prime interest geologically, will not be considered in this effort.
- II. For biological sampling it is assumed that:
 - (A) Particulate material (not solid rock) is of prime interest.
 - (B) Surface material is of prime interest, near subsurface next and deep subsurface least.
 - (C) In the absence of loose particulate matter scraping and abrading of the solid or cemented surfaces is desirable.
 - (D) Comminution of particulate matter is not desirable as it does not increase the biological content of the sampler.
 - (E) Density and size sorting may be desirable.
- III. For geological sampling it is assumed that:
 - (A) Subsurface material or solid rock is preferred to surface material.

- (B) In the absence of loose particulate matter scraping and abrading of solid or cemented surfaces is desirable.
 - (C) Comminution of particulate matter may or may not be desirable depending upon 1) the instrument for which the sample is intended and 2) whether or not it is accompanied by sorting.
 - (D) Density, size, and shape sorting are not desirable.
- IV. Samplers must be simple to enhance reliability; clearly, any kind of sample is better than no sample.
- V. Samplers, if simple, will also be light, small and low powered, thus increasing the possibility of sending multiple samplers for redundancy and/or for covering other modes of sampling.
- VI. Two basic types of samplers will be considered:
- (A) Tethered samplers horizontally deployed some distance from the capsule and employing a 1) flexible tether for random deployment (wire, rubber tube, etc.) or 2) rigid or semi rigid tether (boom, etc.). These will probably be basically surface samplers which may be moved to sweep a larger sampling surface by:
 - (1) Flexible tether
 - (a) Reeling the device in toward the capsule as it samples.
 - (b) Providing locomotion at the sampling end by rolling, crawling, hopping, etc.
 - (2) Rigid tether
 - (a) Mechanical (pre-programmed) motion.
 - (B) Samplers which are fixed to the capsule and deploy essentially vertically (if at all). These are basically subsurface samplers, but may also sample from the surface or the atmosphere. They may be provided with random or programmed motions along the surface to increase area coverage or alternately may employ sweeping devices to bring the sample to the sampler.

- VII. A design goal for the first sampler(s) to be landed on Voyager missions for which 5 pounds has been mentioned might be to include at least one of each of types (A) and (B) since their sampling characteristics are basically different, as indicated below.

BASIC SAMPLER CHARACTERISTICS

TYPE A		TYPE B
Deployed (horizontally from capsule) Sampler		Undeployed (horizontally) (or limited deployment)
Sample size	small	larger
Source of sample	surface to a very limited depth below	surface to an appreciable depth below the surface
Area sampled	may be appreciable	rather limited
Susceptibility to wind	may be so great as completely to preclude obtaining sample	much less affected
Effect of obstructions on sampling capability	would tend to be less	may preclude obtaining any sample

From the above it may be seen that one prime reason for not employing a type (A) sampler exclusively is its vulnerability to wind, and the reason for not employing only a type (B) sampler is the likelihood of its being rendered inoperative by obstructions.

In order to evaluate the sample handling systems conceived under this study phase, JPL developed a performance criteria rating questionnaire. The individual items composing the questionnaire were derived from the rationale which outlines the known and theoretical Martian environment, design goals and system constraints. The initial questionnaire was modified somewhat to include some additional engineering criteria and a numerical rating system was introduced wherein each performance factor was weighted on a 0 to 4 basis. The final sampler performance evaluation criteria list follows.

SAMPLER PERFORMANCE EVALUATION CRITERIA

Overall System Performance

1. Probable Performance in JPL Soil Model 1.
2. Probable Performance in JPL Soil Model 2.

3. Probable Performance in JPL Soil Model 3.
4. Probable Performance in JPL Soil Model 4.
5. Probable Performance in JPL Soil Model 5.
6. Probable Performance in JPL Soil Model 6.
7. Sensitivity to Surface Configuration.
8. Sensitivity to High Wind.
9. Sensitivity to Frozen State of Surface Material.
10. Sensitivity to Precise Sampler Orientation.
11. Sensitivity to Gravity (Orientation and Magnitude).
12. Sensitivity to Atmospheric Pressure or Composition.
13. Sensitivity to Partial or Incomplete Deployment.
14. Sensitivity to Retrieval or Retraction for Acquiring Sample.
15. Ease of Attaining Significant Area Coverage.
16. Ease of Deployment Beyond Landing-Effectuated Zone.
17. Ease of Redeployment for Multiple Sample Coverage.
18. Ability to Discriminate Soil Properties.

Sensitivity of Probable Mechanization to Expected Environment

11. Sensitivity to Sterilization Procedures.
2. Sensitivity to Pre-Launch Ground Handling.
3. Sensitivity to Launch and In-Transit Environment.
4. Sensitivity to Entry and Landing Environment.
5. Sensitivity to Surface Operating Environment:
 - a. Temperature and Temperature Gradients.
 - b. Atmospheric Pressure.
 - c. Atmospheric Composition.
 - d. Wind (Dynamic Pressure).
 - e. Blown Dust.

Quality of the Sample

1. Probable Size of Sample.
2. Probable Upper Limit of Particle Size.
3. Probable Lower Limit of Particle Size.

4. Probable Bias in Sample According to:
 - a. Particle Size (Sorting).
 - b. Density (Sorting).
 - c. Shape (Sorting).
 - d. Composition (Sorting).
 - e. Location (Depth).
5. Sample Modification:
 - a. Mechanical.
 - b. Thermal Degradation.
 - c. Chemical Contamination.

Probable Sampler Physical Characteristics

1. Probable Sampler Size.
2. Probable Sampler Weight.
3. Probable Instantaneous Electrical Power Required.
4. Probable Total Electrical Energy Required (Per Sample).
5. Simplicity Rating.
6. Sensitivity to Close Tolerances of Alignments.

Sampler/Capsule Integration

1. Ease of Storage.
2. Ease of Deployment;
 - a. From Capsule.
 - b. Through Impact Limiter.
3. Probable Capsule Damage;
 - a. Mechanically During Deployment.
 - b. By Vibration or Impact During Operation.
 - c. By Electrical or Radio Interference.
 - d. By Chemical Contamination (Electrical Overheat, etc.).
4. Ease of Sample Extraction from Sampler.
5. Probable Sampler Failure Diagnosis by On-Board Imaging System.
6. Probable Sampler Failure Repairability or Correctability.

2.2 SAMPLE HANDLING SYSTEM CONCEPTS

The development of ideas for sample handling systems was initiated by a Technical Specialty Seminar held at Aeronutronic on 16 February 1966. The seminar was conducted for the purpose of informing a wide spectrum of in-house individual disciplinary specialists about the ABL sampling program and to elicit from them suggestions and comments concerning sampling and handling methods from the viewpoint of their various specialties. The attendees consisted of responsible senior personnel representing the scientific and engineering disciplines of biology, geology, space physics, mechanical engineering, engineering mechanics and instrumentation. Many suggestions and basic concept variations and modifications resulted from this meeting.

A standardized format was developed for recording the more pertinent sample handling systems concepts generated during the study phase of the program. All of the concepts presented to JPL for initial review and evaluation were drawn and described on this form. The original conceptual designs, recorded on this standard format which illustrates the concept by means of a three-dimensional sketch and provides explanatory notes including supporting requirements (deployment mechanism, sample transport device and sample processing method), operational characteristics (weight, volume, power, operational simplicity and operation), estimating quality of sample (size range, bias and sample degradation characteristics), overall performance estimate (re soil models and Martian environmental properties), and reference data (source of original idea), are reproduced in Appendix B.

A total of 50 sample handling system concepts were generated during the course of the conceptual design phase. Since by definition, a sample handling system consists of sample acquisition, transporting, processing and transferring devices, each concept will include specific examples of each type of device. Each type of device can be classified according to its primary mechanical principle and the concepts listed according to their design conformity to a mechanical principle or as an alternate configuration of a basic design. Tables 2-II to 2-V break down the subsystem components per this type of classification and designate the components used per concept by the concept identification numbers appearing in Appendix B as follows:

Table 2-II Sampler Deployment and Retrieval Concepts

Table 2-III Sample Acquisition Concepts

Table 2-IV Sample Transport Concepts

Table 2-V Sample Processing and Transfer Concepts.

TABLE 2-II

LIST OF SAMPLER DEPLOYMENT AND RETRIEVAL CONCEPTS

No.	Utilized by Concept No.	Mechanical Principle		Basic Concept	Alternate Configuration
		Deployment:			
1	2A-2, 2A-3, 6A, 6B, 8D, 10D	Vertically deployed tube	Single tube	Fixed for deployment and retrieval only	
2	4B, 8B-1, 8B-2			Rotating	
3				Oscillating	
4	3A, 4A		Double tube	Combination, fixed and rotating	
5	4D, 5D			Combination, rotating and oscillating	
6	4C, 4C-2			Combination, rotating and sliding	
7	2A-2	Vertically deployed wireline	Gravity deployment of sampler		
8		Vertically deployed impact	Gravity drop		
9	6A		High velocity impact		
10	2A-1, 2A-3, 5A, 7A, 8D, 10G	Horizontally deployed boom	Telescoping		
11	2A-1, 2A-3, 5A, 7A, 8C		Furlable		
12	8A		Rigid hinged		
13	1A, 2B, 10B	Ballistic deployment	Reaction jets		
14	2B, 5B-1, 5C, 7D, 10B		Springs		
15	2D, 3B, 5B-2	Crawling sampler head	Flexible tube		
16			Wireline		

TABLE 2-II (Contd.)

No.	Utilized by Concept No.	Mechanical Principle	Basic Concept	Alternate Configuration
		<u>Retrieval:*</u>		
1	6A	Vertical tube, as above	Single tube	Retrieves sampler and sample
2			Double tube	Retrieves sampler and sample
3				Retrieves sample only (transport device)
4		Vertical wireline, as above	Retrieves sampler and sample	
5	4D,8D,10B	Boom, as above	Telescoping (may have internal transport tube) and sample	Single tube, retrieves sampler and sample
6	2A-3,7A			Dual tube, expands sampling area
7	5A,8C		Furlable (may trans- port as well as deploy)	Retrieves sampler and sample
8	2A-3,7A			Expands sampling area
9	8A		Rigid, hinged	Expands sampling area
10	2B,5B-1,5C,7D	Wireline	Retrieves sampler and sample	
11	1A,2D,5B-2,10B	Flexible tube	By reeling, increases sampled area and re- trieves sampler, if desired.	

*Retrieval applies to retrieval of the sample acquisition device (sampler and only to the sample in case the sample is retrieved with the sampler.)

Explanation: No. refers to concepts generated; Concept No. refers to preliminary design data presented in Appendix A.

TABLE 2-III

LIST OF SAMPLE ACQUISITION CONCEPTS

No.	* Concept No.	Mechanical Principle	Basic Concept	Alternate Configuration
1	* 1A	Aerosolizing head	Litton configuration	
2	*	Aerosolizing head with accessories	Superpressure configuration	Air curtain seal
3	* 2A-2			Flexible, vertically deployed seal
4	* 2A-3			Labyrinth seal
5	* 2A-1			Force balance to control lift
6	*			Lateral control for mobility
7	* 2B		Fixed external brushes or scrapers	
8	*		Powered external brushes	
9	* 2D		Brushes on hole-seeking head	
10	* 2A-1, 2A-3		Supersonic surface-eroding jet	
11	* 3A	Helical screws	Rigid, vertically deployed	
12	* 3B		Flexible, horizontally deployed	
13	* 4A	Soil augers and drills	Tubular with flexible flutes	
14	* 4B		Conical with collecting shoulder	
15	* 4C-1		Tubular with separate external flutes, internal tube	Flexible fingers to hold sample
16	* 4C-2			Spin dumped, cone-shaped cup to hold sample
17	* 4D		Rotary impact drill	

TABLE 2-III (Contd.)

No.	* Concept No.	Mechanical Principle	Basic Concept	Alternate Configuration
18	*	5A Scoops and scrapers	Boom deployed back hoe	
19	*	5B-1	Dragline retrieved clamshell scoops	Radial scoops closed on command
20	*	5B-2		Crawler activated by opening and closing action of flutes
21	*			Opening and closing scraping action
22	*		Clamshell scoops with pneumatic transport	Closed compartment for super-pressure development
23	*	5C	Semi-passive clamshell, hinged tetrahedron	
24	*	5D	Vertical rotary scoop	
25	*	10B	Bellows actuated multiple clamshell	
Penetrating plugs or core punches; implanting energy may be: High velocity impact Reaction from explosively driven mass				
26	*	6A Rotary impact	Cylindrical plug with spring fingers	
27	*	6B Static loading, with or without vibration	Array of small diameter tubular plugs	
28	*	7A Surface abrasion devices with or without pneumatic action	Rotating wire brush, cylindrical or shaped	

TABLE 2-III (Contd.)

No.	* Concept No.	Mechanical Principle	Basic Concept	Alternate Configuration
29	*		Rotating abrasive wheel, cylindrical or shaped	
30	* 7C		Reciprocating or oscillating brush or abrasive surface	
31	* 7D		Dragline retrieved brush or abrasive surface	
32	* 10G		Continuous wire brush	
33	* 8A	Abrasive sieves	Horizontally deployed cylinder	
34	* 8B-1		Vertically deployed abrasive sieve	Sample mechanically dumped
35	* 8B-2		Boom deployed abrasive sphere	Sample spin dumped
36	* 8C		Horizontally deployed disk-shaped abrasive sieve with centrifuge particle collector	
37	* 8D			
38		Adhesive coated surface	Coated collecting surfaces	Deployed string or tape
39				Boom deployed coated conforming surface
40				Coated collecting surface used with brush or abrasive
41				Coated collecting surface for windborne particles
42		Bulk application of adhesive material to surface	Collection of consolidated sample	

TABLE 2-III (Contd.)

<u>No.</u>	<u>* Concept No.</u>	<u>Mechanical Principle</u>	<u>Basic Concept</u>	<u>Alternate Configuration</u>
43		Application of liquid to surface	Liquid jets for erosion and transport	
44			Liquid applied and frozen to consolidate sample	
45				Bore hole wall stabilization for deep sampling
46	* 10A	Expanding-contracting slotted tubes	Inflated slotted tube	
47	*		Mechanically actuated slotted tube	
48		Electrostatic or magnetic pick-up	Electrostatic attraction of particles	
49			Magnetic gathering of particles	Native magnetic particles only
50	* 10D			Utilizing iron collecting elements with or without agitation

 50 40 32 TOTALS

Explanation: No. refers to ideas generated; * concepts presented to JPL;
Concept No. refers to preliminary designs in Appendix A.

TABLE 2-IV

LIST OF SAMPLE TRANSPORT CONCEPTS

<u>No.</u>	<u>Utilized by Concept No.</u>	<u>Mechanical Principle</u>	<u>Basic Concept</u>	<u>Alternate Configuration</u>
1	5C, 8D	Mechanical transport of sampler and sample	Boom	Telescoping
2	5C, 8D			Furlable
3	5B-1, 7D		Wireline	
4	4D, 5A, 7A, 8B-1, 8C	Mechanical transport of sample only	Upraised boom for grav- ity drop transport	
5	2B		Wireline retrieved sample collection chamber	
6			Adhesive string or tape	
7		Vertical mechanical elevators	Telescoping tubes or ascending internal collectors	
8	4D		Nonrotating oscillating saw-tooth ledges	
9	4C-1		Piston released plugs	Plunger releasing flexible- fingers
10	6A			Cylindrical sleeve releasing flexible fingers
11	6B			Grating activating multiple pistons
12	3A, 3B, 10D	Tubular mechanical conveyors	Helical screw	
13	10G		Endless conveyor	Brush
14	10B			Bellows

TABLE 2-IV (Contd.)

<u>No.</u>	<u>Utilized by Concept No.</u>	<u>Mechanical Principle</u>	<u>Basic Concept</u>	<u>Alternate Configuration</u>
15	4A,4B,4C-2, 8B-2	Centrifuge devices	Spin dumped ascending collecting cup	
16	8D		External ring collector	
17			Rotating oscillating sawtooth ledges	
18	7A	Aerosol transport	Telescoping tube mounted within boom	
19	2A-2		Flexible tube	Single- or multi-stage blower
20	1A,2A-1,2A-3, 2D,5B-2,8A,8C			Aspirator or inductor jet
21	7C	Pneumatic tape or tube	Rolled-up tape	
22	10A		Reeled-in slotted tube	
23	10D	Magnetic	Entrapment by iron collecting elements	

Note: Some concepts employ dual or multiple sample transport principles,
e.g., 4D (vibrating conveyor and gravity drop).

TABLE 2-V

LIST OF SAMPLE PROCESSING AND TRANSFER CONCEPTS

No.	<u>Utilized in Concept No.</u>	<u>Mechanical Principle</u> <u>Processing (Sizing):</u>	<u>Basic Concept</u>	<u>Alternate</u> <u>Configurations</u>
1	8B-1, 8B-2, 8C, 8D, 10A, 10B, 10G, 4B, 4D, 5D, 6B, 7A, 7C, 7D, 8A	Mechanical grading	Mechanical grading during acquisition of sample	
2	3A, 3B, 4B, 5D, 7A		Mechanical grading during transport of sample	
3	4A, 4C-2, 4D		Sieves	
4	1A, 2B	Aerosol jet action	During acquisition of sample	
5	2A-1, 2A-2, 2A-3, 2D, 5B-1, 5B-2, 7C, 8C, 10G	Pneumatic grading	During acquisition and/or transport of sample	
2-18	4C-1, 5A, 5C, 6A	No sample processing		
		<u>Transfer:</u>		
1	5A, 5C, 5D, 6A, 6B, 7A, 7D, 8B-1, 8B-2, 8C, 10A, 10B, 3A, 3B, 4A, 4B, 4C-1, 4C-2, 4D	Gravity transfer	Gravity dump and drop system within capsule	
2	2A-1, 2A-2, 7A, 7C, 8C, 10A, 10B	Pneumatic transfer	Omnidirectional pneumatic tubes	Single- or multi-stage blowers
3	1A, 2A-3, 2B, 2D, 5B-1, 5B-2, 8A, 10G			Aspirator or inductor jet
4	10D	Mechanical conveyors	Omnidirectional helical screws	
5			Vertical vibrating saw-tooth edges	Nonrotating
6				Rotating

Note: Some concepts employ multiple sample processing or transfer concepts or devices.

The total number of concepts deemed worthy of preliminary design treatment numbered 40 (denoted by asterisks on Table 2-III). A total of 32 preliminary conceptual designs were submitted to JPL for evaluation and rating. These 32 designs are presented on standardized formats in Appendix B.

The preliminary design review was held at JPL on 22 March 1966. In attendance were:

<u>JPL</u>		<u>Aeronutronic</u>
G. Bastien	E. A. Howard	W. H. Bachle
C. F. Campen	H. R. Lawrence	D. H. Garber
J. A. Dunne	D. B. Nash	T. W. Neumann
A. G. Ford	G. A. Soffen	
H. Ford	J. L. Taylor	
G. M. Hotz		

At this meeting Aeronutronic was requested to make a preliminary recommendation of the sample handling systems deemed most worthy of continued investigation. Aeronutronic personnel recommended four classes of sampler concepts because one or more of the subsystem components within each concept showed promise of producing significant improvement in the performance of unmanned automated sampling systems. These recommended concepts included:

- | <u>Vertically Deployed</u> | <u>Horizontally Deployed</u> |
|--|---|
| 1. Superpressure aerosolizing head employing:
a) Flexible deployable seal,
b) Supersonic eroding jets,
c) Surface agitation by rotating wire brush. | 3. Boom deployed rotating abrading head employing:
a) Spherical abrading sieve and/or
b) Wire brush |
| 2. Conical abrading sieves (considering several alternate configurations). | 4. Self-deploying pneumatic sampling tube employing:
a) Pressurized batch or continuous particle transport, and
b) Crawling clamshell head, or
c) Powered burrowing aerosolizing head. |

On 8 April 1966, a program review meeting was held at Aeronutronic, JPL representatives, G. M. Hotz and E. A. Howard, listed the preliminary concepts itemized on Table 2-VI as having the highest evaluation scores on the performance criteria rating questionnaires judged by the JPL concept evaluation team.

TABLE 2-VI

JPL EVALUATION OF AERONUTRONIC SAMPLING SYSTEM CONCEPTS

Concept No.	Description	Score (Based on 0-4 Point Rating Basis)	
		Type A Horizontally Deployed	Type B Vertically Deployed
*1A	Aerosolizing jet (Litton)	686	VIII
2A-2	Superpressure aerosolizing head		768 VI
*3A	Rigid helical screw		816 IV
3B	Flexible helical screw	787	V
*5A	Back hoe (Surveyor)	727	VII
5B-2	Clamshell scoops	777	VI
5D	Rotating scoop/ vibratory feed		862 I
6A	Plug sampler		827 III
7A	Rotating wire brush	796	IV
*8A	Cylindrical abrading sieve	844	II
8B-1	Conical abrading sieve		830 II
8B-2	As above with spin dump		769 V
8C	Spherical abrading sieve	880	I
8D	Abrading disk	843	III

*Under development by JPL, in-house or contract.

The specific samplers under development by JPL were then omitted from the list and, in view of the recommendations of Aeronutronic, the following 8 preliminary concepts were scheduled to be carried through preliminary design stages:

Concept No.	Description	Relative Rank (1 - Highest)	
		Type A	Type B
2A-2	Superpressure aerosolizing head		5
5D	Rotating scoop/vibratory feed		1
6A	Plug sampler		3
7A	Rotating wire brush	3	
8B-1	Conical abrading sieve		2
8B-2	As above, spin dumped		4
8C	Spherical abrading sieve	1	
8D	Abrading disk	2	

Another design review was held at Aeronutronic on 4 May 1966. The following personnel were in attendance:

<u>JPL</u>	<u>Aeronutronic</u>
G. Bastien	W. H. Bachle
A. G. Ford	D. H. Garber
G. M. Hotz	W. Hostetler
E. A. Howard	T. W. Neumann

As a result of this design review, JPL indicated by letter dated 13 May 1966, that the following four preliminary designs be pursued in greater detail:

Concept No.	Description	Type
5D	Rotating scoop/vibratory feed	B Vertical deployment
7A	Rotating wire brush	A Horizontal
8B-2	Conical abrading sieve/spin dump	B Vertical
8C	Spherical abrading sieve	A Horizontal deployment

Continued parametric studies and detailed design factors led Aeronutronic personnel to recommend two principle concepts for prototype design and development at the final design review meeting held at Aeronutronic on 15 June 1966. This meeting was attended by G. M. Hotz and E. A. Howard of JPL, and W. H. Bachle, D. H. Garber, W. Hostetler and R. S. Kraemer from Aeronutronic. The following two principle concepts proposed by Aeronutronic conform to contractual requirements, for one is of Type A (horizontally deployed) and the other, Type B (vertically deployed).

Type A	Concept 7A	Rotating wire brush
Type B	Concept 8B-2	Spin dumped conical abrading sieve

Concurrence with this recommendation by Aeronutronic personnel was received in a letter from JPL dated 20 June 1966. Coincident with the receipt of this letter, the breadboard test fixture fabrication program was launched.

2.3 PARAMETRIC SUBSYSTEMS STUDIES

During the Conceptual Design Phase, engineering design studies were concurrently carried forth on three accessory subsystems: boom deployment devices, supersonic nozzle configurations, and pneumatic motors.

2.3.1 DEPLOYABLE BOOMS

When considering a system for deploying a sampler to some horizontally removed spot on the surface from a landed payload, mechanisms which can be compactly stowed and yet achieve some reasonable length are desired. Mechanisms which are suggested are booms of various types such as expandable trusses, inflatable structures, telescoping cylinders, and the more sophisticated furlable tapes which can be uncoiled from a drum to form a tube. For this study the telescoping cylinders and the furlable tape appeared to offer the best prospects for developing a deployable boom that could achieve some reasonable horizontal range for a minimum of weight with design simplicity and operational reliability. The parametric study was conducted in order to obtain a quantitative assessment of some of the characteristics of a tubular boom.

The analysis progressed in steps. First, the mechanism of extension was ignored so that the geometric and physical variables could be investigated in terms of strength and weight. These give basic boom characteristics. The assumption is made that the wall thickness of the telescoping boom is thin enough so that the results obtained for a constant diameter tube are sufficiently accurate. This assumption is valid only so long as the wall thickness is small with respect to the tube diameter and the number of telescoping elements are small. The diameter in this case is considered to be the average diameter. Second, the differences between telescoping booms and furlable booms are explored to determine their relative merit and finally some preliminary design aspects for the furlable boom are deployed.

The first consideration to be made in a structure such as a deployable boom is the type of material that should be used and its compatibility with dry heat sterilization, biological contamination requirements, and structural strength. The basic materials considered in this study are identified in Table 2-VII along with their pertinent physical characteristics.

TABLE 2-VII

PHYSICAL PROPERTIES OF REPRESENTATIVE BOOM MATERIALS⁹

<u>Material</u>	<u>E</u>	<u>σ_c</u>	<u>E/σ</u>	<u>ρ</u>
Phenolic Fiberglass*	4.81×10^6	47500	101	.065
Beryllium/Copper	19×10^6 *	120000	158	.290
Magnesium AZ31B	6.5×10^6	12000	540	.065
Steel PH 15-7 Mo	30×10^6	185000	162	.290

*Reference 8

These materials do not represent all possible candidate materials but cover a sufficiently broad spectrum to develop meaningful parametric data.

The first comparison which can be easily made is to determine the specific boom length for the various material densities as a function of tube diameter and wall thickness. Specific length is here defined to mean the length of boom which can be constructed per pound of material used. This data is presented in the family of curves shown in Figure 2-1 for magnesium and fiberglass and Figure 2-2 for steel and beryllium/copper. In examining these figures it is seen that very long tube lengths can be achieved for very low weights; however, structural strength limitations will probably require that the heavier wall tubes will be used as well as larger diameters. Boom lengths up to 20 feet long are possible for a weight expenditure of one to five pounds. The specific length is seen to decrease rapidly with increased tube diameter but eventually approach a minimum value asymptotically. In the size range of probable interest with tubes one to two inches in diameter and wall thickness of .010 to .020 inches, it is seen that the proper selection of tube diameter is not as critical as is that of wall thickness and material density.

In order to assess the strength limitations for a deployable boom, some form of loading must be assumed. This loading consists of the distributed loading of the boom along its length due to the structure and the concentrated load that the boom is expected to deploy. In this case the concentrated load is a soil sampler which for the sake of conservatism is assumed to be 5 pounds earth weight. The assumed loading with the associated shear, moment, slope, and deflection diagrams are shown in Figure 2-3. The parametric data were calculated using earth gravity rather than Mars on the premise that the sampler system must be capable of operating on earth in order to be tested and checked out prior to flight. This will provide a safety factor in the flight hardware which

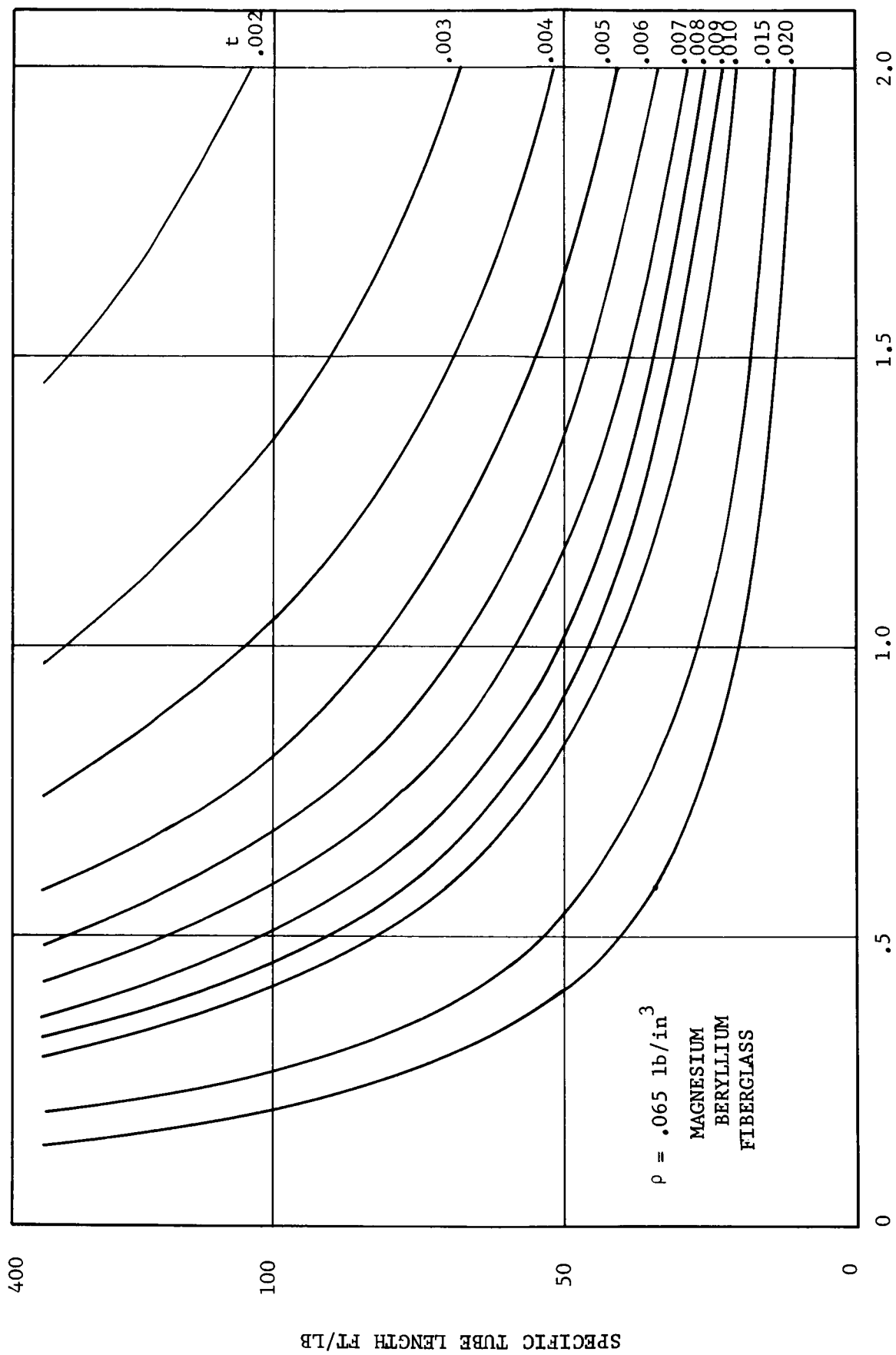


FIGURE 2-1 TUBE LENGTH PER POUND OF MATERIAL

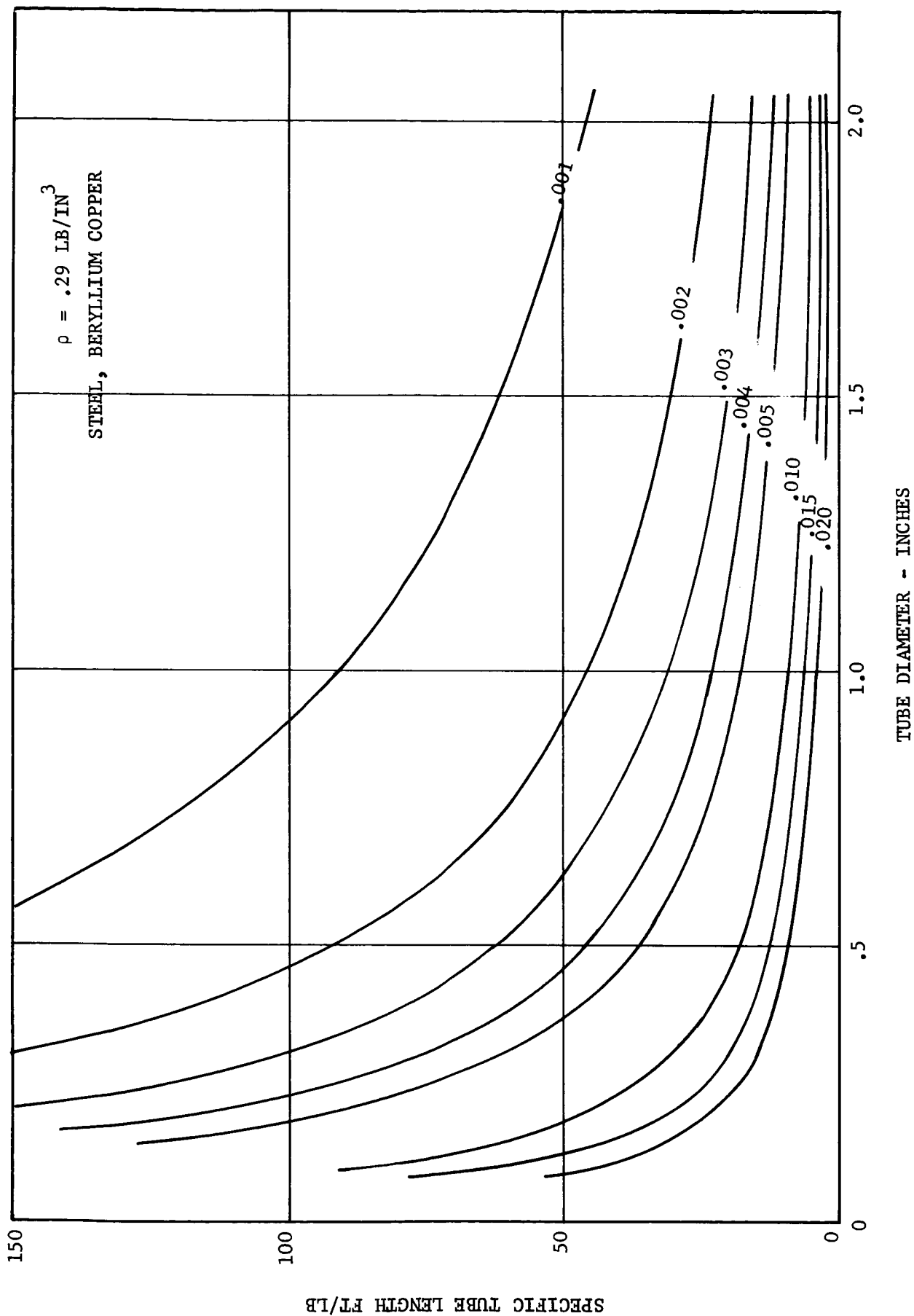


FIGURE 2-2 TUBE LENGTH PER POUND OF MATERIAL

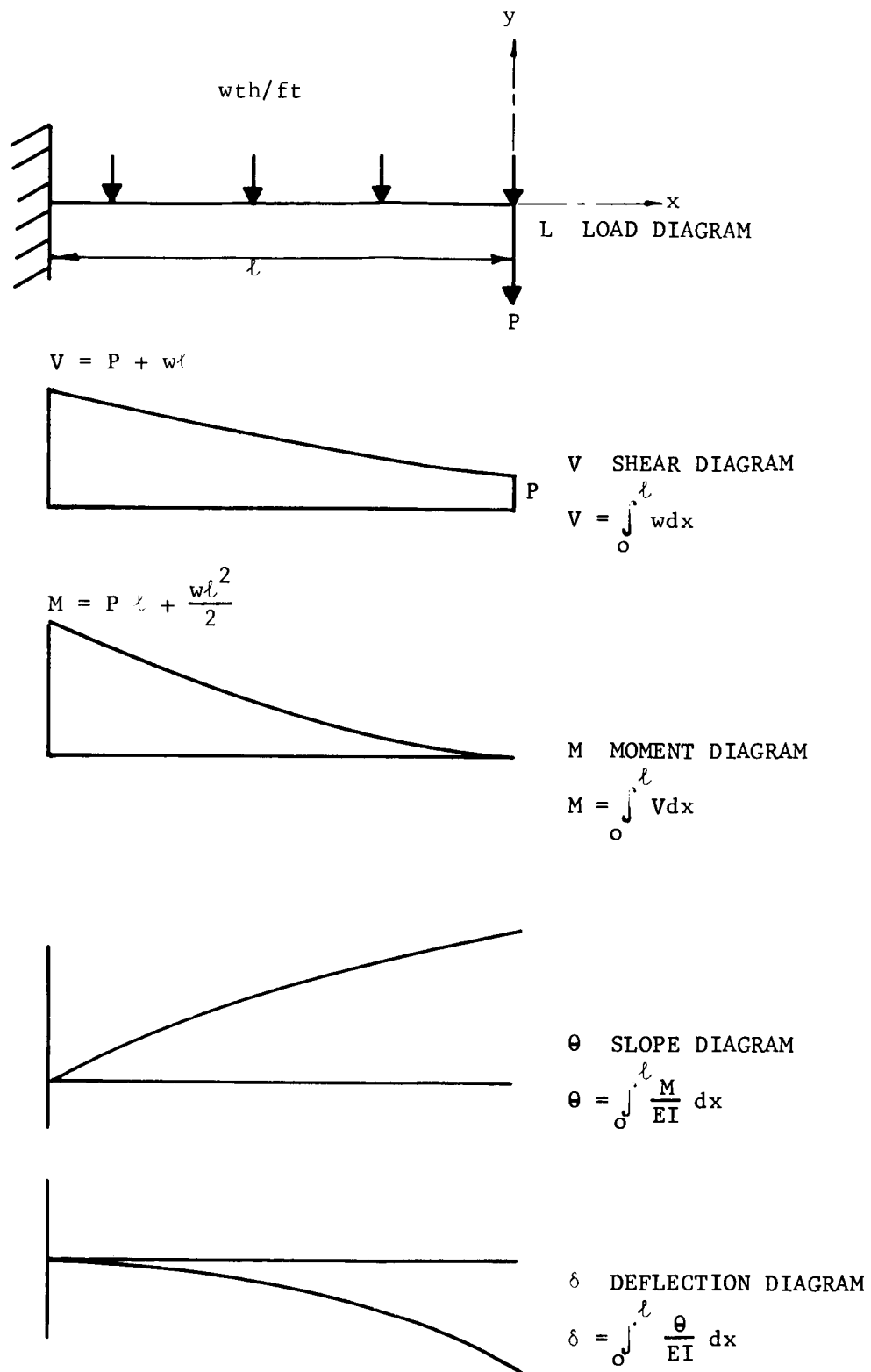


FIGURE 2-3 BOOM LOADING USED IN ANALYSES

is proportional to the ratio of the gravitational acceleration for earth to Mars which is 2.67. The bending stress is given by $\sigma = M_c/I$ and the moment at any point x is given by $M_x = P_x + W_x^2/2$. Substituting the maximum value of the moment, which occurs at $x = l$, and the allowable bending stress σ_c , an expression can be obtained for the maximum allowable boom length. This expression is

$$l = \frac{-P}{\pi d t \rho} \pm \sqrt{\frac{\sigma_c d}{\rho} + \left(\frac{P}{\pi d t \rho}\right)^2}$$

which was used to calculate the allowable boom length in terms of the allowable bending stress and boom geometry.

In addition to determining the allowable boom length based on the allowable bending stress of the material, the allowable buckling stress must be considered for thin walled tubing. The critical bending moment is given by the expression $M_{cr} = k(E/1 - \mu^2)rt^2$ taken from reference 1. The constant k can vary from .72 to 1.14; however, the theoretical value of .99 can be used for long tubes with sufficient accuracy. Families of curves relating deployed length as a function of tube diameter for various wall thicknesses were calculated based on the material allowable bending stress. These are shown for phenolic fiberglass in Figure 2-4, for beryllium/copper in Figure 2-5, and for PH 15-7 Mo stainless steel in Figure 2-6. The point at which the buckling or crippling stress becomes equal to the allowable bending stress of the material is determined by setting the ratio of the critical crippling moment to the allowable bending moment equal to one. This yields the expression

$$\frac{M_{cr}}{M} = k \frac{E}{1 - \mu^2} rt^2 \cdot \frac{c}{\sigma I} = \frac{k}{\pi(1 - \mu^2)} \frac{Et}{\sigma r} = 1.$$

Solving for the tube diameter yields the expression $d = \frac{kt}{2\pi(1 - \mu^2)} \frac{E}{\sigma}$.

The diameter calculated in this manner identifies the point at which the crippling stress is equal to the allowable material bending stress. This can be referred to as the critical crippling diameter. These critical diameters are tabulated in Table 2-VIII.

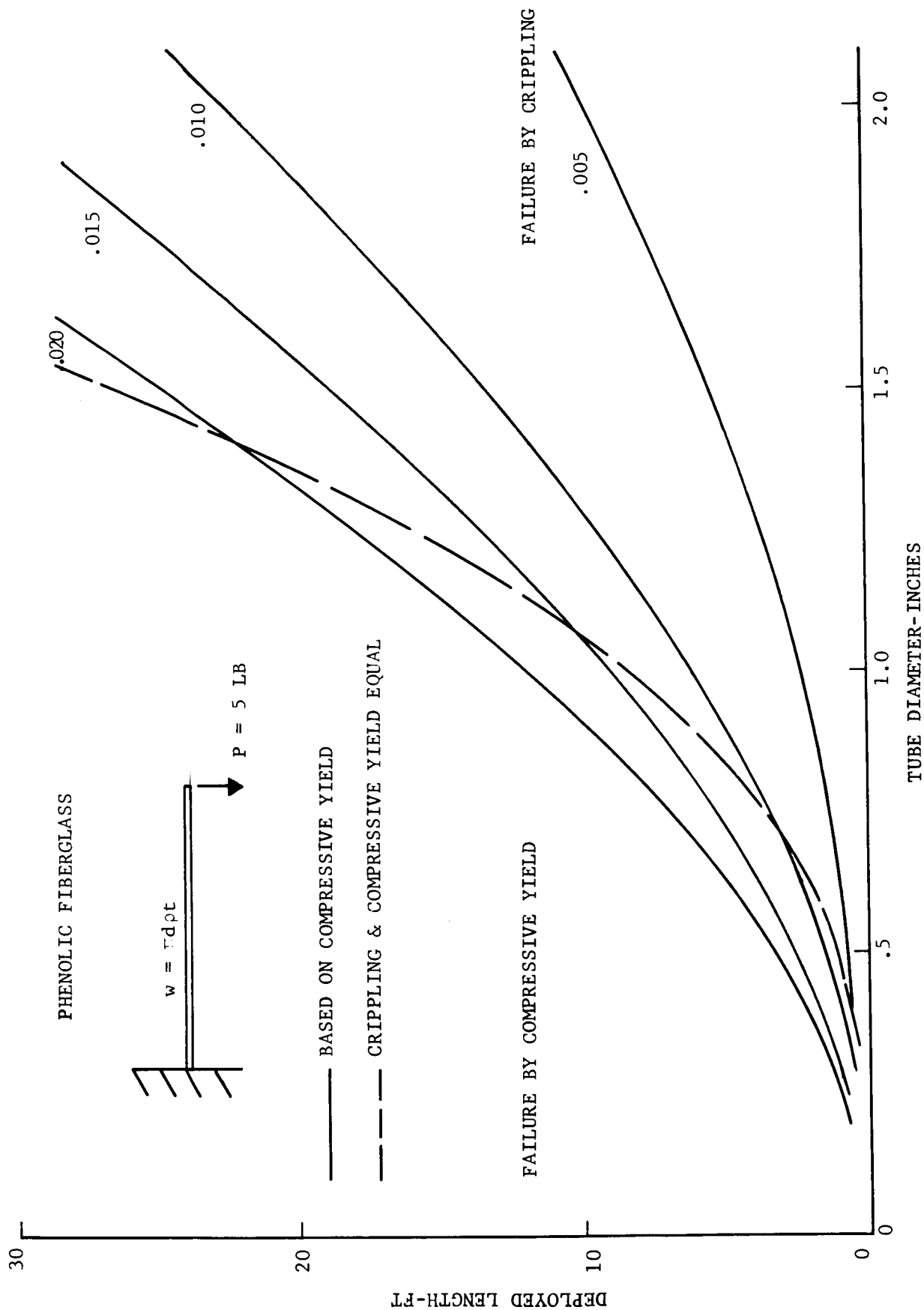


FIGURE 2-4 DEPLOYABLE BOOM LENGTH

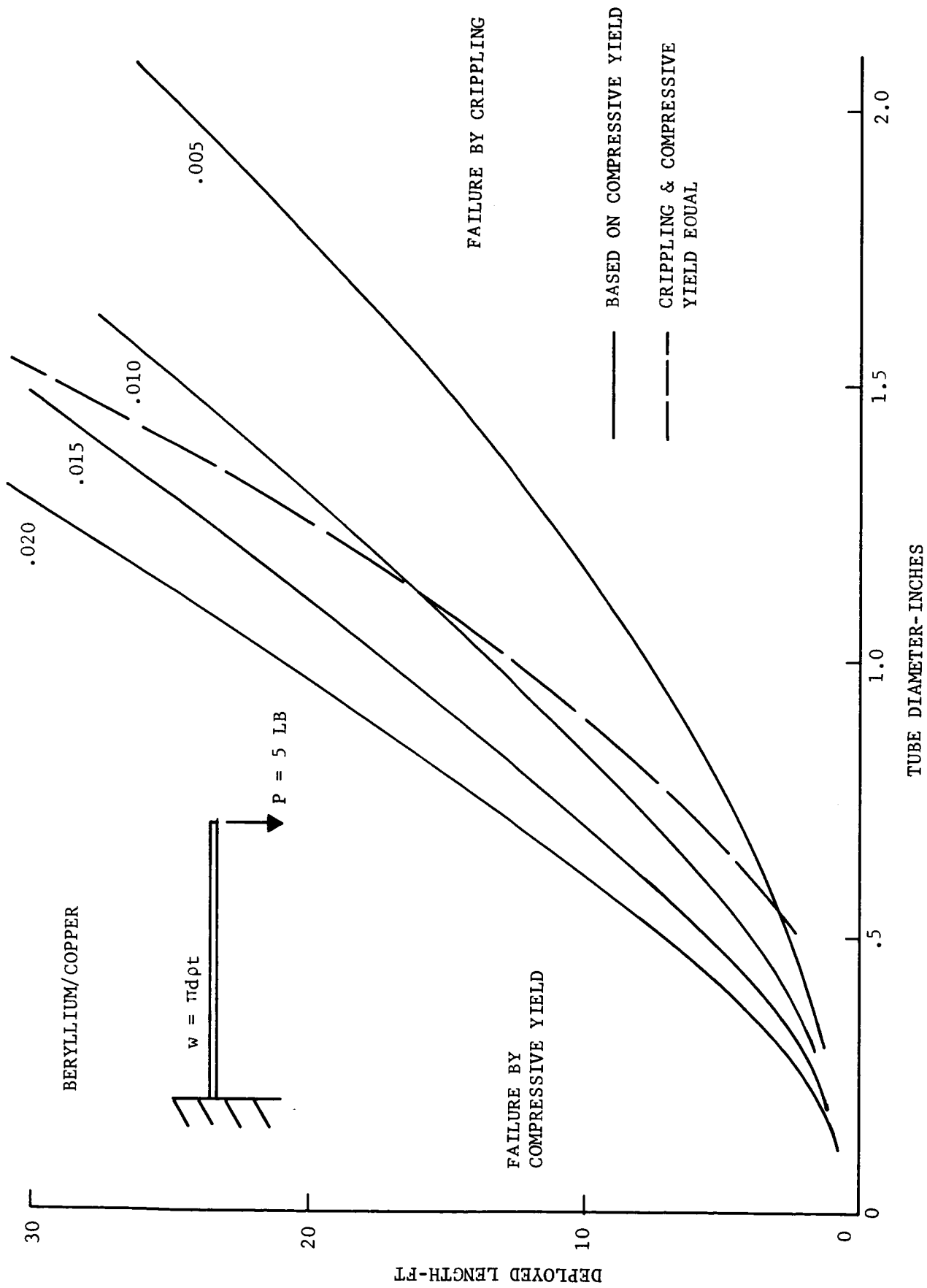


FIGURE 2-5 DEPLOYABLE BOOM LENGTH

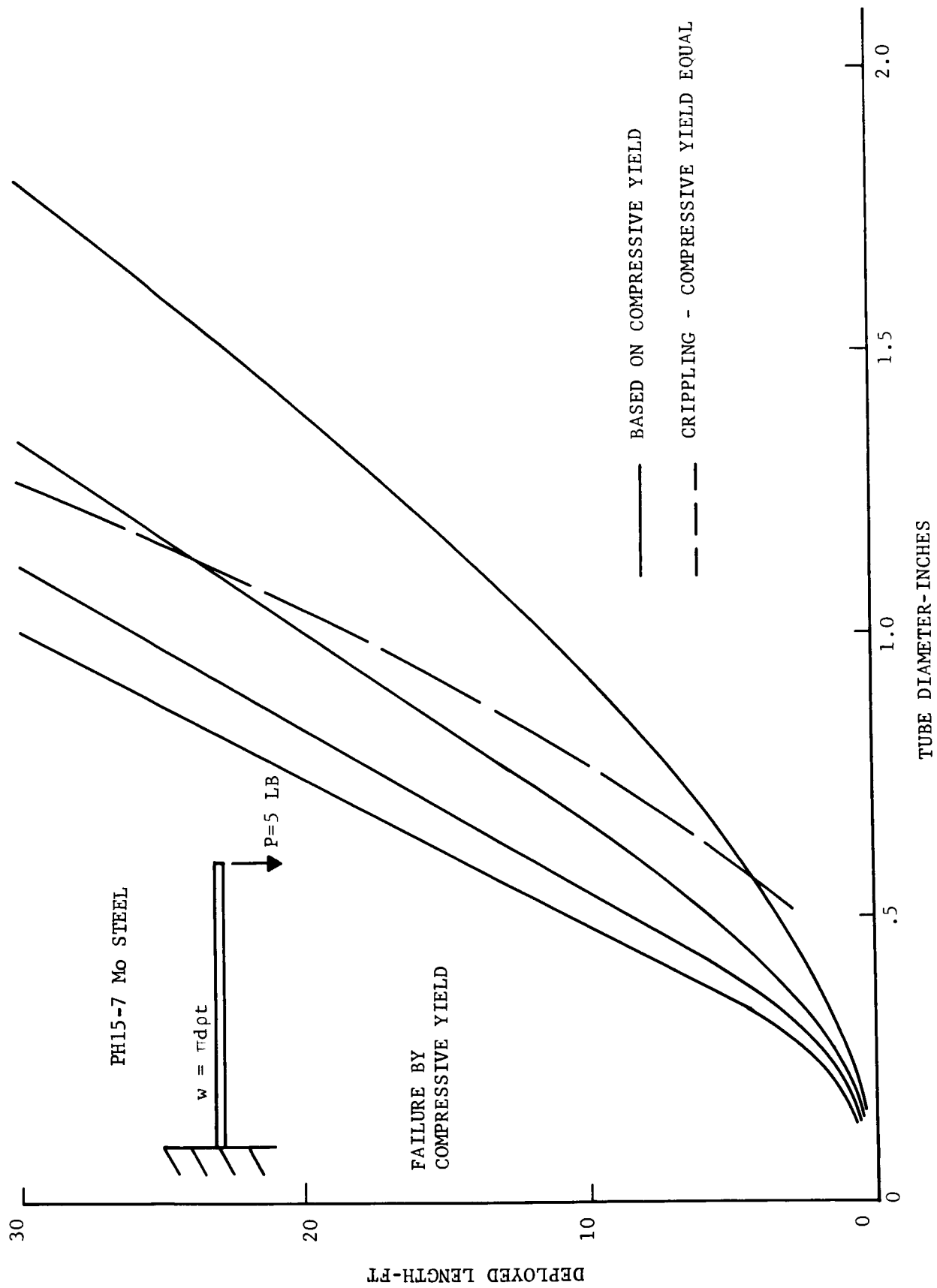


FIGURE 2-6 DEPLOYABLE BOOM LENGTH

TABLE 2-VIII

CRITICAL CRIPPLING DIAMETER FOR TUBES IN BENDING

	t = .005	t = .010	t = .015	t = .020
Phenolic Fiberglass	.35	.7	1.06	1.41
Beryllium-Copper	.55	1.11	1.66	2.21
PH15-7Mo Steel	.57	1.13	1.70	2.26

These values are cross plotted on Figures 2-4 through 2-6 as the dashed line. For the purposes of this study, it was assumed that useful boom designs should avoid the region of failure by crippling in the interest of achieving higher structural reliability. From this parametric data it is seen that boom deployment lengths of 20 feet are possible with realistic combinations of wall thickness and tube diameter. Using the critical crippling points on each curve and the specific tube length data, a final plot relating minimum boom weight as a function of deployed length is derived and presented in Figure 2-7. From these data it is seen that fiberglass is most desirable as a boom material based on strength characteristics. It should be remembered at this point that these represent minimum weights based on a strength criteria. Operational constraints may very well impose a deflection criteria which may be more severe, particularly for long deployment lengths.

A consideration of the effect of surface winds on the allowable deployed length should be made. The term $\pi d t \rho$ is the distributed load, w , per unit length. The expression previously derived for allowable length can be modified to account for wind loads by adding the wind load to the distributed load. The expression for the allowable boom length now becomes

$$l = \frac{-P}{w + \bar{w}} \pm \sqrt{\frac{\pi d^2 t \sigma}{2(w + \bar{w})} + \left(\frac{P}{w + \bar{w}}\right)^2},$$

where \bar{w} is the added load due to wind. This assumes that the wind is acting in the same plane as the body loads of the boom, which means the wind would be blowing vertically downward. This is not a probable condition but will yield conservative results while simplifying the calculations. A maximum wind velocity of 500 feet per second was assumed. This value should be sufficiently high to cover the most severe gust conditions that have been predicted. The model 3 or 10 millibar atmosphere given in TN D-2525 was used in these calculations. This results in a dynamic pressure of 50.5 pounds per square foot which is equivalent to a wind

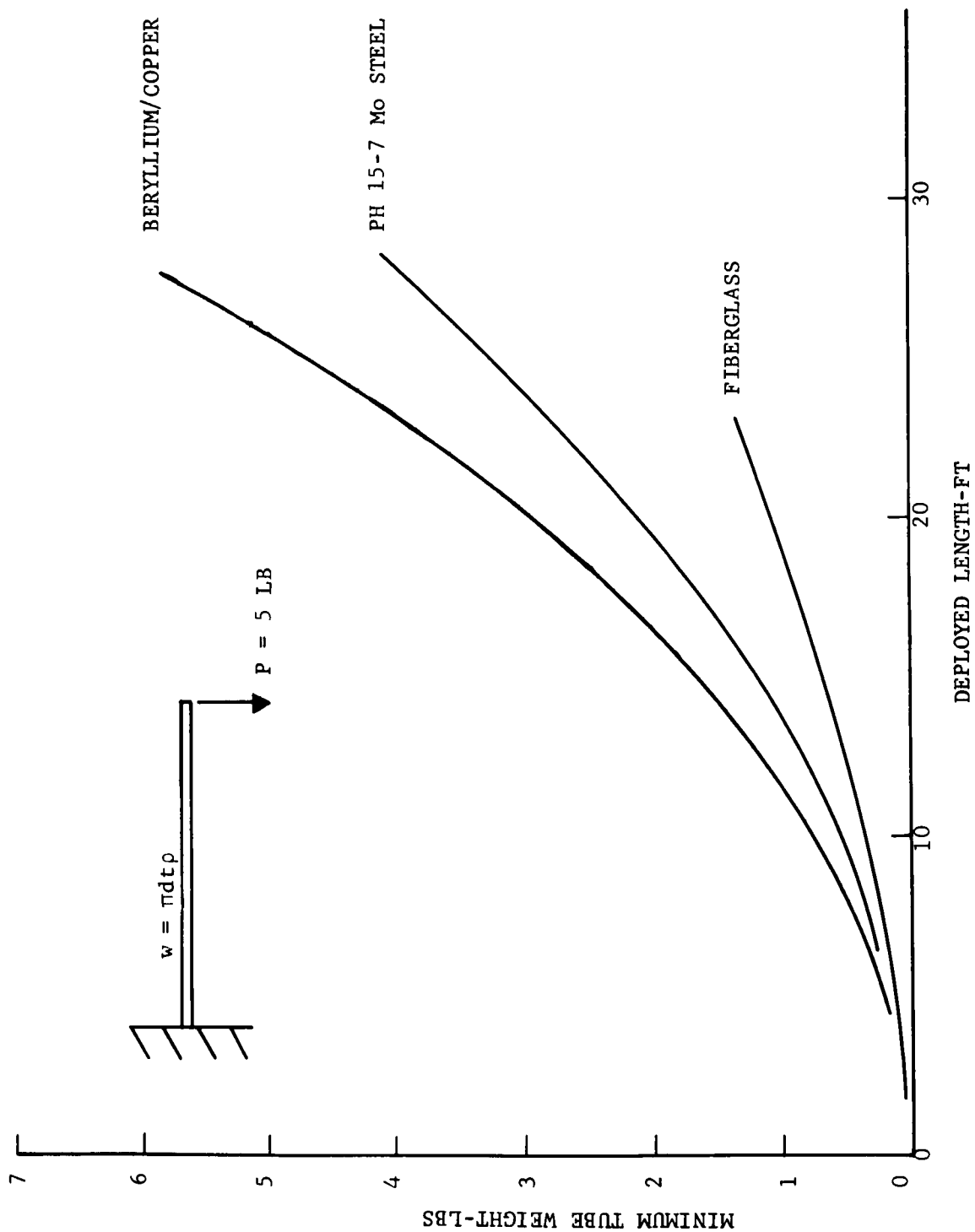


FIGURE 2-7 MINIMUM TUBE WEIGHT FOR EXTENDED BOOM

velocity in an earth atmosphere of 67 feet per second or 46 miles per hour. This is converted into an equivalent loading on the boom by calculating the drag force on a cylinder. At moderately high Reynolds numbers, the drag coefficient for a cylinder is 1.2. From this the effective wind load is calculated using the expression

$$\bar{w} = \frac{1}{2} \rho S C_D v^2 = q C_D d$$

per unit length of the boom. The allowable tube length in a wind is shown in Figure 2-8 for phenolic fiberglass, Figure 2-9 for beryllium/copper, and Figure 2-10 for PH 15-7 Mo steel. Comparing these figures with Figures 2-4 through 2-6, it is seen that the allowable deployed length is considerably reduced from approximately 20 feet to 10 feet for the same tube geometries. It should be noted here that this analysis is based on a static loading which will probably not be true in the real case. A cylinder in a wind can shed vortices alternately from one side and then the other which will act as a dynamic forcing function on the boom. Should resonance occur, this could result in destruction of the boom. The dynamic action of the boom in a high wind or in gusts can only be verified by wind tunnel testing. The severe wind condition calculated here is probably unrealistic as an actual design criteria; however, it does serve to identify the upper limit of the effect of wind on the deployable length of the boom. The degradation in deployed length was not as severe as might have been expected for such a high wind velocity.

In either a telescoping boom or a furlable boom design, torsional strength will suffer because of the sliding joints in a telescoping boom and because of the open tube section of the furlable boom. The allowable torsional moment for an allowable shear stress of 40,000 psi is shown in Figure 2-11. In the range of tube sizes from one to two inches in diameter, the allowable torque varies from 1200 inch pounds up to 5000 inch pounds for an .020 wall thickness. For most surface sampling systems the strength requirement will probably not exceed one percent of these values. Thus, it is reasonable to expect that the cylindrical segments of a telescoping boom can be keyed together to provide sufficient torsional strength. Likewise, restoring this degree of continuity to a furlable boom should be feasible. In fact, some designs incorporating interlocking teeth along the open edge of the tube have been made by others.

As mentioned previously, the bending deflections for a highly optimized boom could become the most severe design criteria. The tubular boom deflections were calculated for earth gravity conditions using a one pound weight on the end of the boom. Families of curves for various tube diameters and wall thickness were generated. The deflection characteristics for phenolic fiberglass is shown in Figure 2-12, for beryllium/copper in Figure 2-13, and for PH 15-7 Mo steel in Figure 2-14. It is seen that the deflection increases rapidly as the deployed length increases.

PHENOLIC FIBERGLASS
WIND VELOCITY - 500 FPS

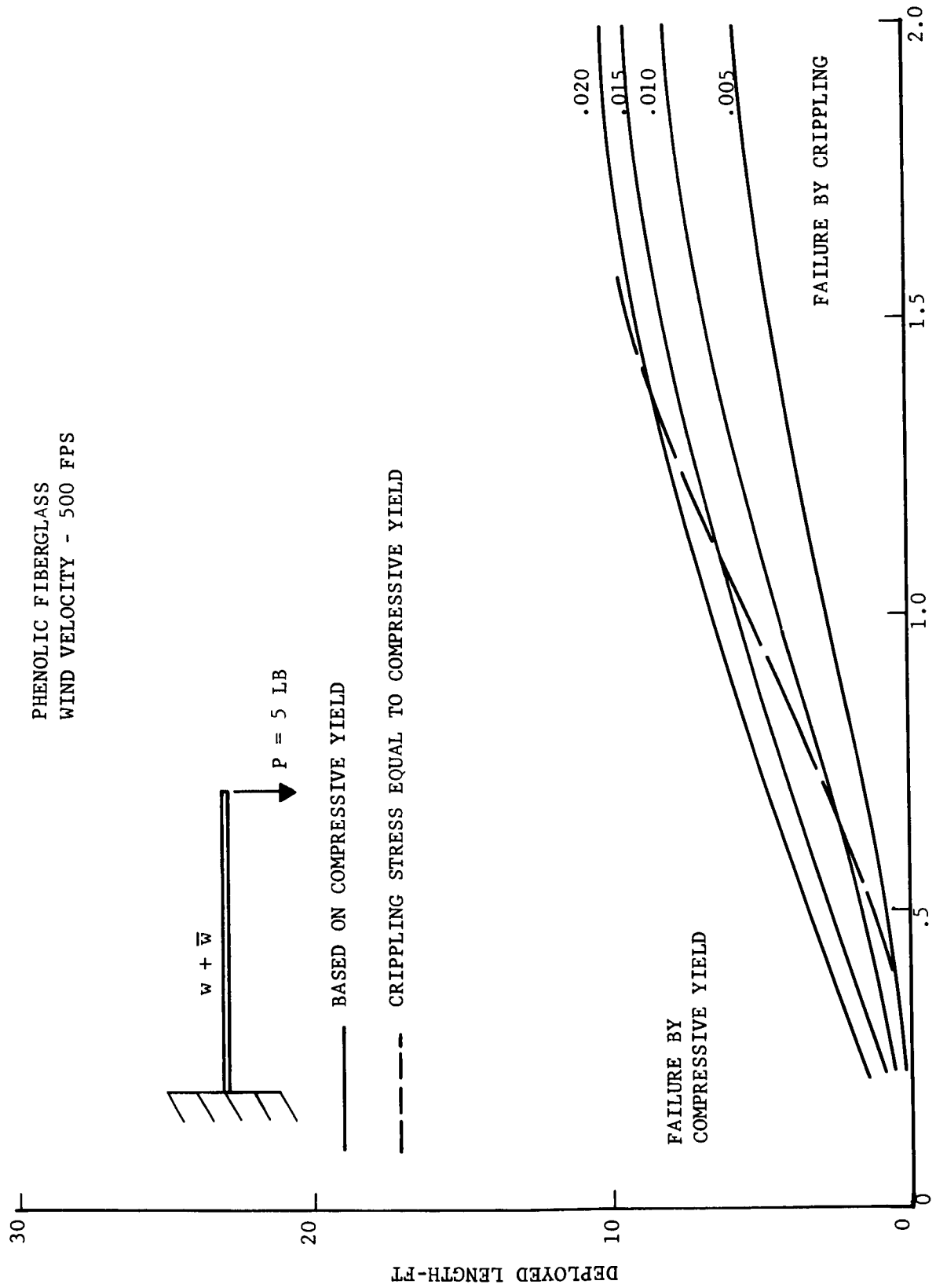


FIGURE 2-8 DEPLOYABLE BOOM LENGTH IN WIND

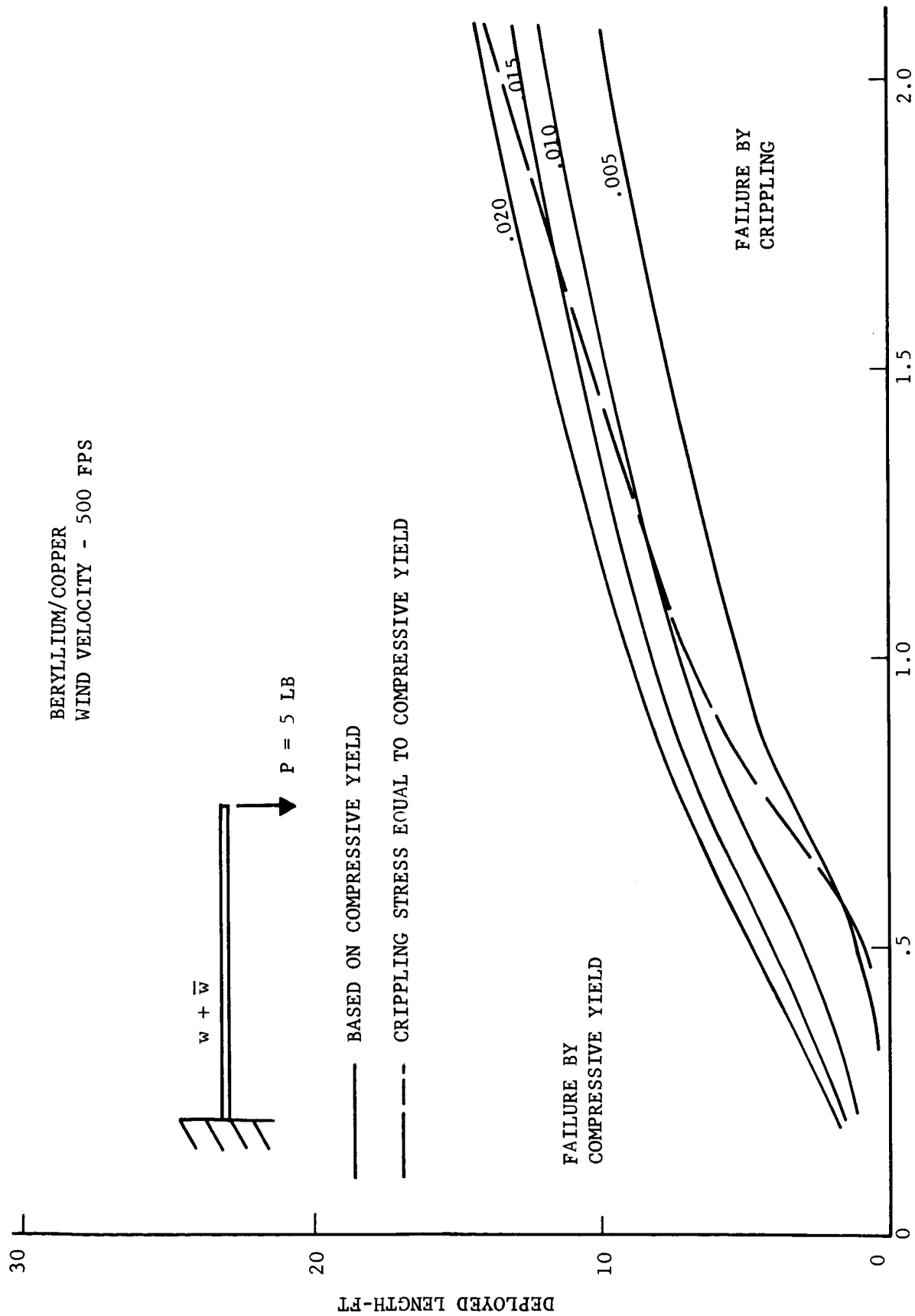
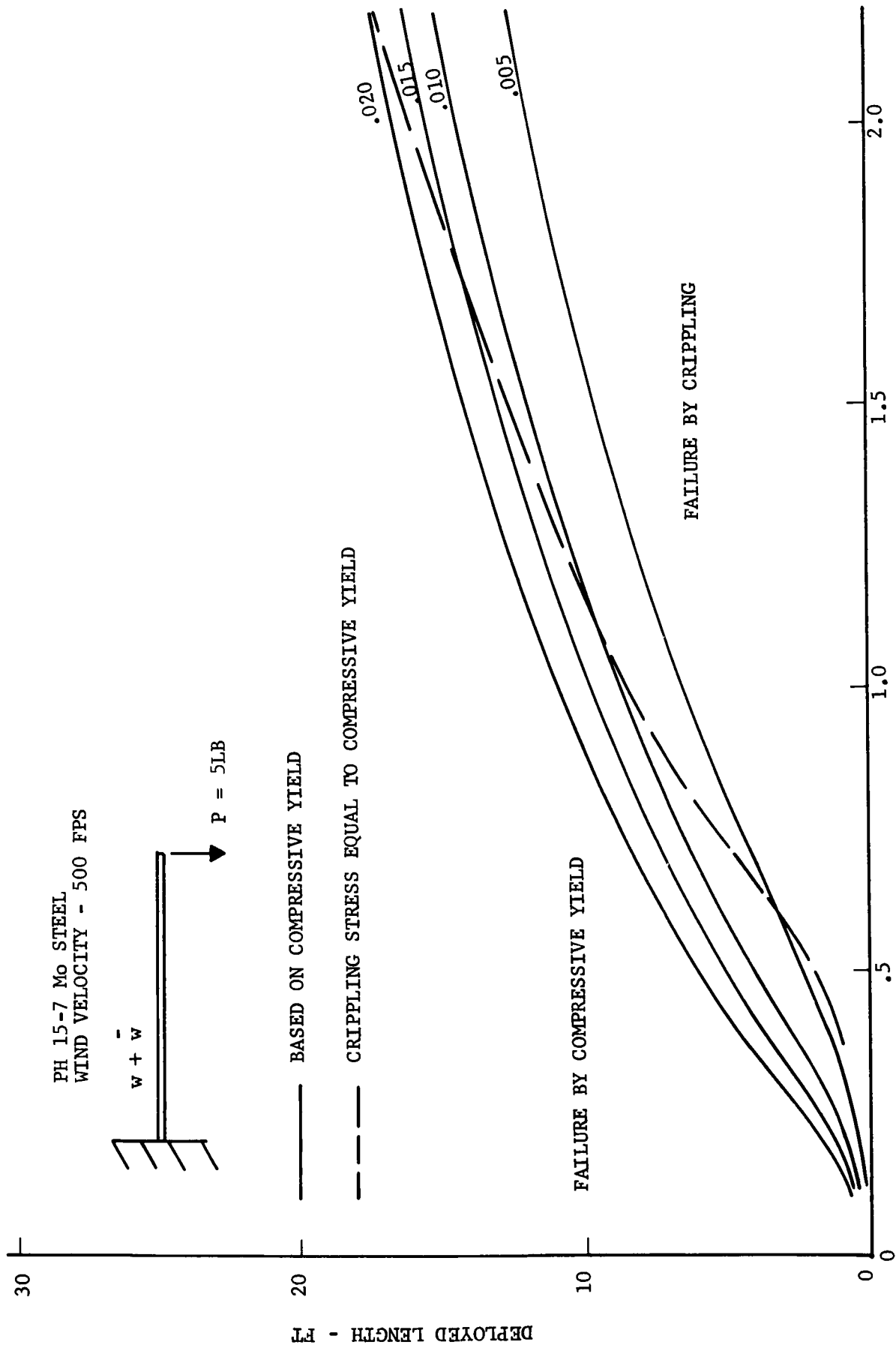


FIGURE 2-9 DEPLOYABLE BOOM LENGTH IN WIND



TUBE DIAMETER - INCHES

FIGURE 2-10 DEPLOYABLE BOOM LENGTH IN WIND

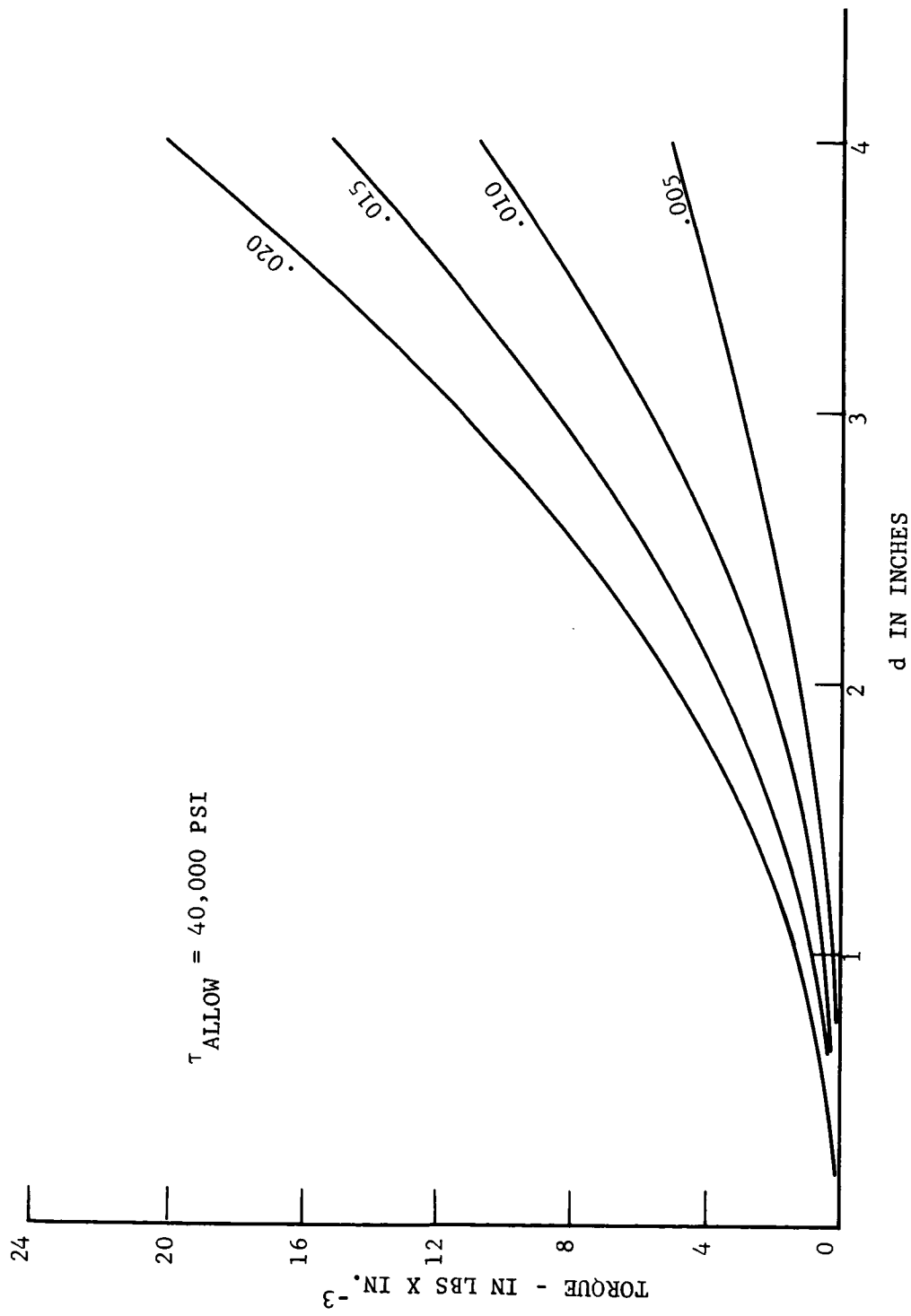


FIGURE 2-11 TORSIONAL STRENGTH AS A FUNCTION OF TUBE WALL THICKNESS

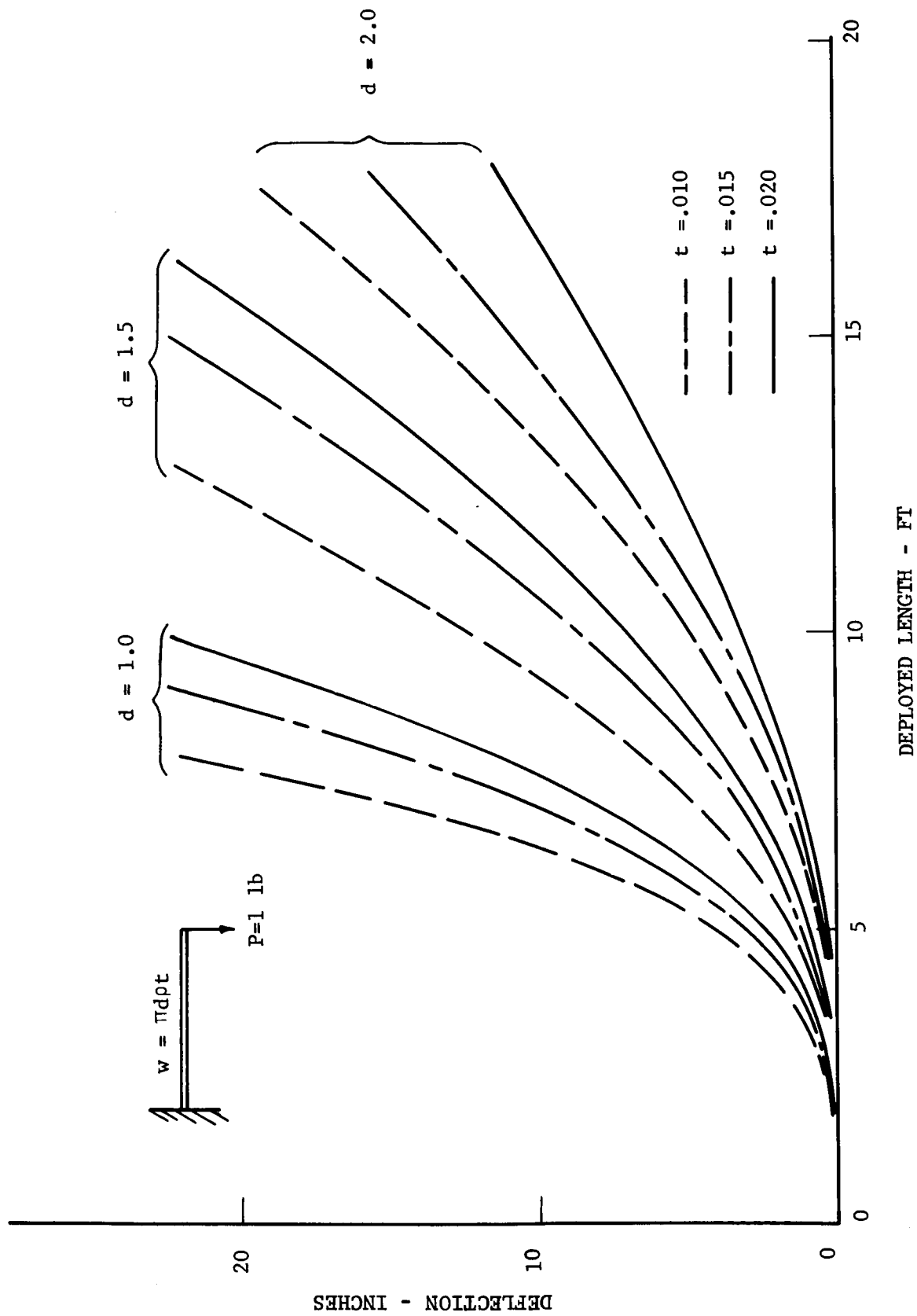


FIGURE 2-12 DEFLECTION CHARACTERISTICS OF A FIBERGLASS BOOM

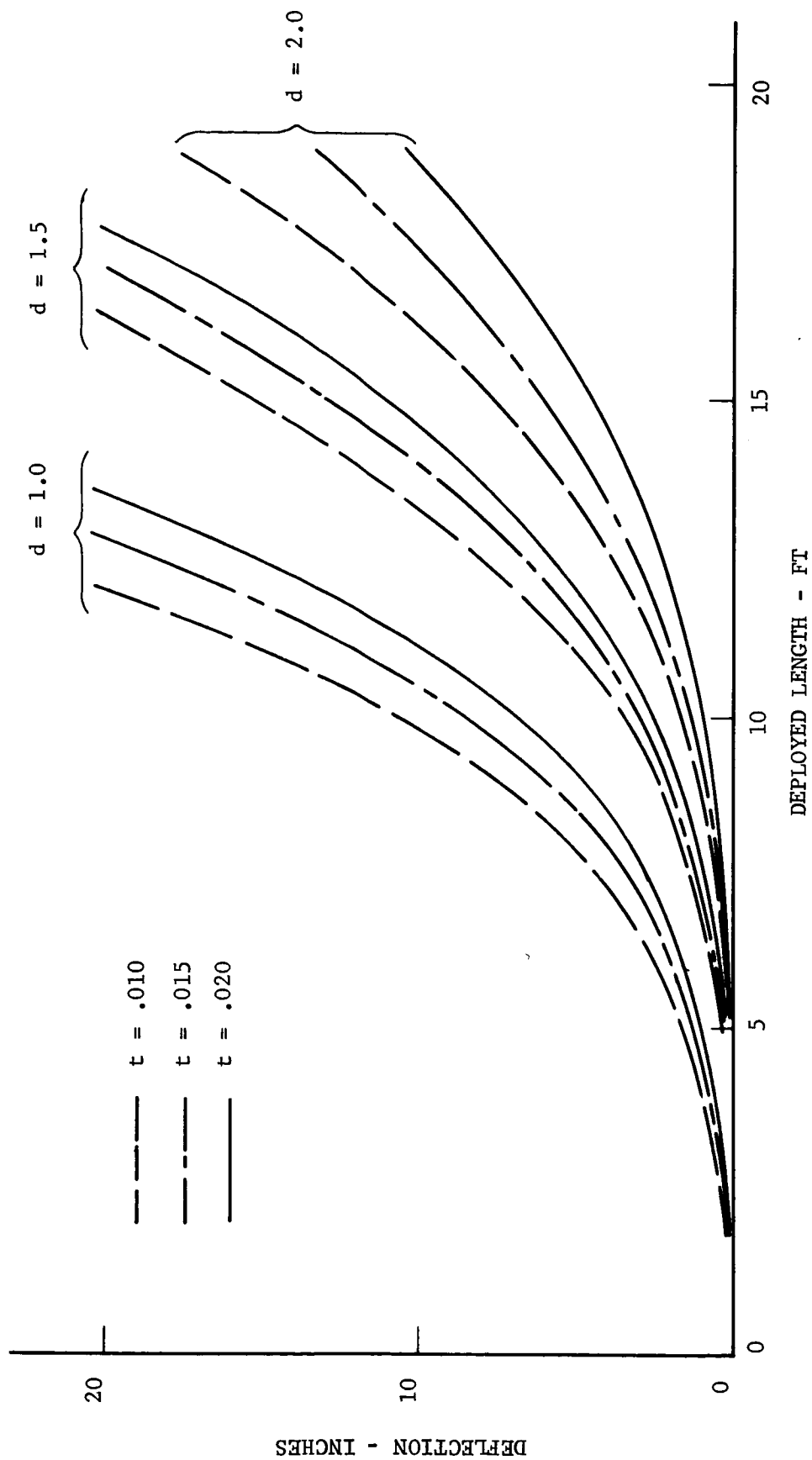


FIGURE 2-13 DEFLECTION CHARACTERISTICS OF A BERYLLIUM/COPPER BOOM

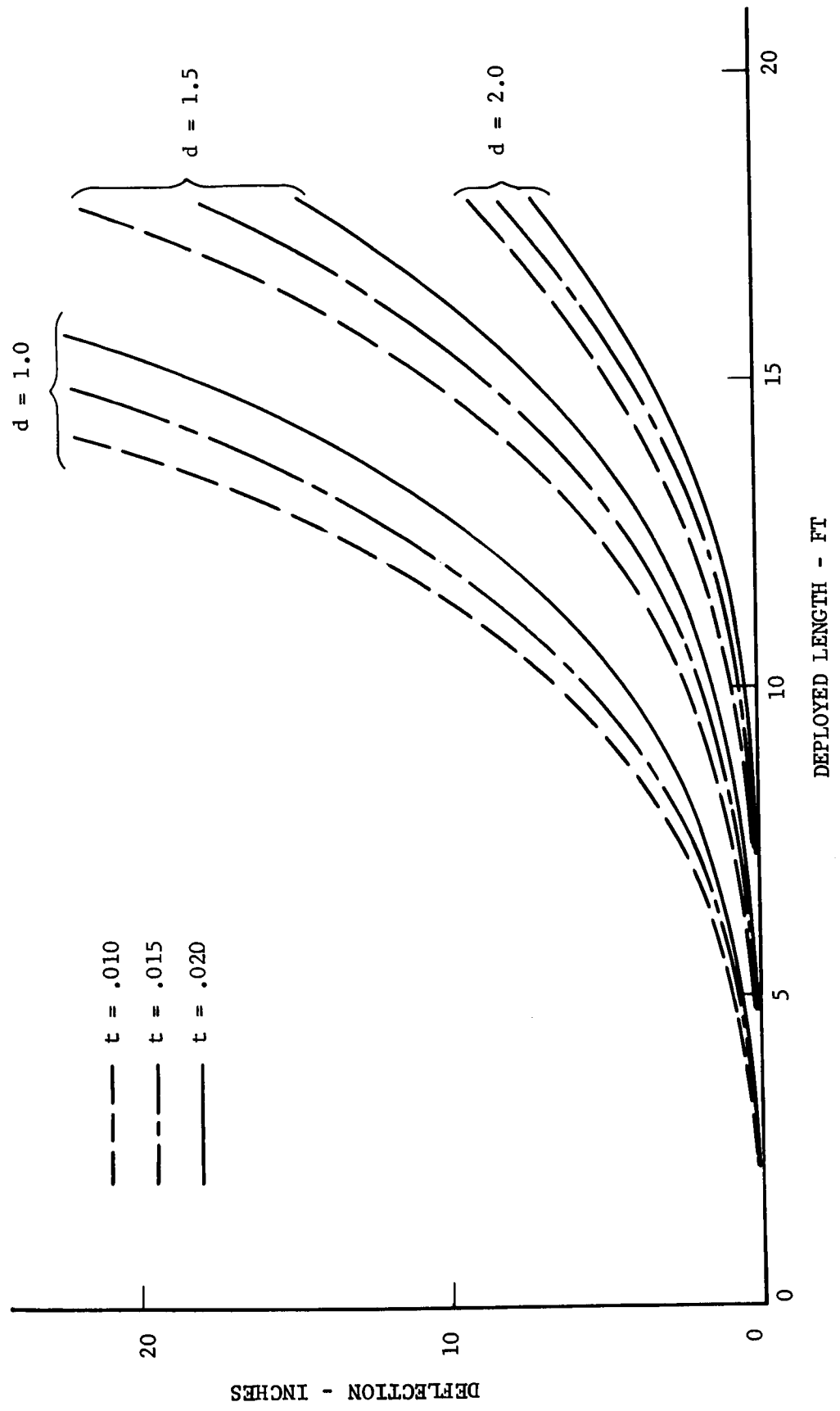


FIGURE 2-14 DEFLECTION CHARACTERISTICS OF A STEEL BOOM

Since the amount of deflection that can be tolerated is directly related to the constraints imposed by the intended operation of the boom, it is impossible to say what deflection is acceptable in a parametric study. Intuitively, a maximum deflection of 10 inches would appear to be a reasonable limit. If this is applied as a criteria, it is seen that the phenolic fiberglass boom is limited to a deployed length of 10 to 15 feet and the beryllium/copper and steel is limited to a length of 20 feet. It should be borne in mind that the magnitude of the deflection can be reduced at the expense of increased weight and stress level. The deflections that would be experienced on Mars would be reduced from those experienced on earth by the ratio of Mar's gravitational acceleration to earth's gravitational acceleration which is about .38. The minimum boom weight as a function of deployed length was determined for steel, fiberglass, and beryllium/copper and is shown in Figure 2-15. A limiting deflection of 10 inches was assumed when a concentrated load of one earth pound is applied at the tip of the boom. These minimum weights should be viewed with some reservations. One is that this weight is the weight of the tube only. Additional weight will be required to support the end of the boom, for fittings, and for deployment mechanisms. The other reservation is that many of these minimum weights are achieved with unrealistically thin tubes. These weights will probably not be achievable due to limits imposed by fabrication methods; however, this analysis establishes the feasibility of achieving a horizontal range of 20 feet with a deployable boom for a weight under 5 pounds.

A deployable boom design can be achieved by using either telescoping tube elements or furlable tapes. The relative advantages and disadvantages are listed in Table 2-IX.

TABLE 2-IX
RELATIVE MERIT OF DEPLOYABLE BOOM TYPES

<u>Telescoping Cylinder</u>	<u>Furlable Boom</u>
<u>Advantages:</u> <ol style="list-style-type: none"> 1. Higher strength because of closed tube section. 2. Can be deployed pneumatically. 	<u>Advantages:</u> <ol style="list-style-type: none"> 1. Eliminates multiple parts and sliding joints. 2. Easily deployed and retrieved by motor-driven system.
<u>Disadvantages:</u> <ol style="list-style-type: none"> 1. Two-way action is more complex. 2. Weight penalty at sliding joints. 3. Reliability at sliding joints. 	<u>Disadvantages:</u> <ol style="list-style-type: none"> 1. Lower strength because of open section. Wall thickness limited. 2. Volumetric packing efficiency may be less.

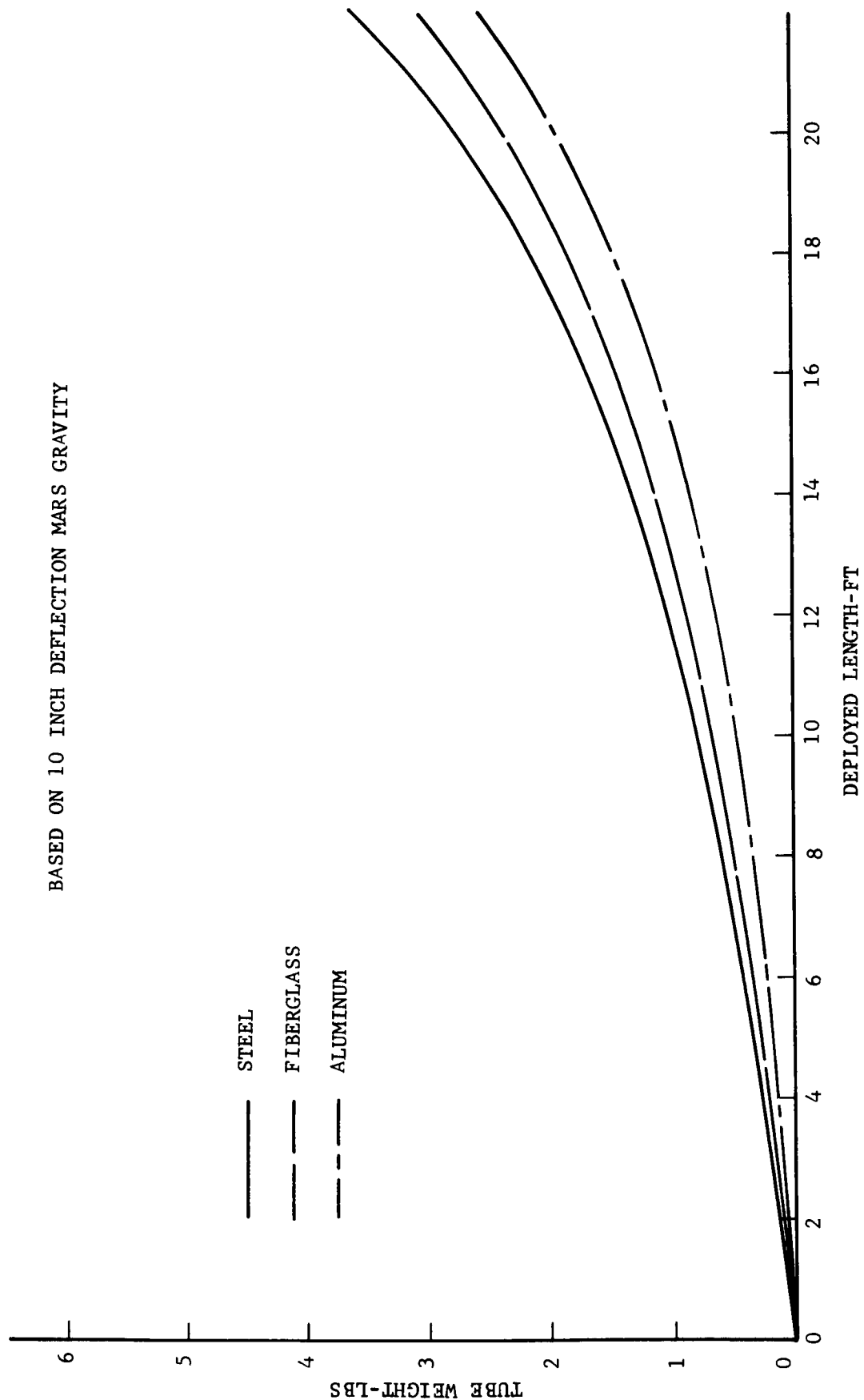


FIGURE 2-15 MINIMUM BOOM WEIGHT BASED ON DEFLECTIONS

The furlable tape boom seems to have a qualitative advantage from the standpoint of simplicity and reliability if low strength can be tolerated. If a high strength boom is desired the telescoping boom must be used.

Some of the design features of a high strength furlable tape boom were evaluated and will be discussed in the following paragraphs. Such a boom is conceived as a tape preformed to a round tubular shape slit along one edge. In storage the tube is flattened and wound on a drum. In this operation the deformations must remain in the elastic range; i.e., no yielding shall occur. The drum is driven by a motor to control deployment and retraction. It should be noted that strain energy is stored during retraction which is available to aid in deploying the boom.

The fundamental variables involved in predicting the allowable geometric proportions are modulus of elasticity E , tube diameter d , wall thickness t , and storage drum diameter D . The conditions which exist in first flattening and then rolling the tape on a drum are shown in Figure 2-16.

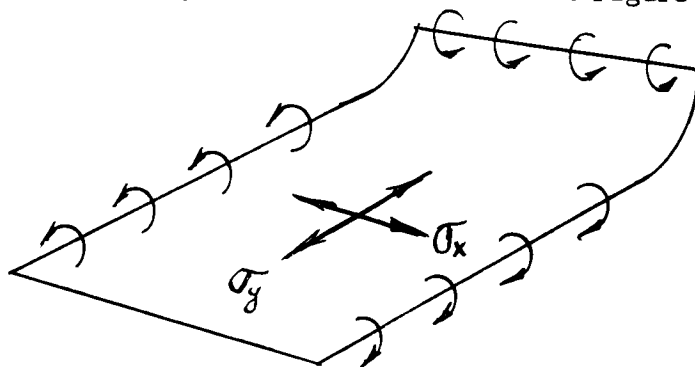


FIGURE 2-16. TAPE FLATTENING AND ROLLING CONDITIONS

The flattening can be accomplished by applying a moment M_f along the edge of the tape. A moment M_D applied to the end of the tape rolls it onto the drum. The stresses acting in the tape are transverse stresses σ_x and longitudinal stresses σ_y . The stress due to flattening the tube is given by

$$\sigma_x = \frac{\epsilon_x E}{1 - \mu^2} \quad (1)$$

and

$$\sigma_y = \frac{\mu \epsilon_x E}{1 - \mu^2} \quad (2)$$

These stresses are tensile on the surface which is the inside of the tube and compressive for the surface which is the outside of the tube. The stress due to rolling the tube on a drum is given by

$$\sigma_y = \frac{\epsilon_y E}{1 - \mu^2} \quad (3)$$

and

$$\sigma_x = \frac{\mu \epsilon_y E}{1 - \mu^2} \quad (4)$$

From reference 4, the strains due to bending in each direction are related to the tape geometry by

$$\epsilon_x = \frac{t}{2R_T} = \frac{t}{d} \quad (5)$$

and

$$\epsilon_y = \frac{t}{2R_D} = \frac{t}{D} \quad (6)$$

The terms R_T and R_D are the initial tube radius and drum radius respectively. Two different sets of conditions for combining the stresses given in equations (1) and (3) can be encountered depending on how the tape is wound on the drum. When the tape is forward wound, the surface which was the inside of the tube becomes the inside of the roll on the drum. When the tape is backward wound the surface which was the inside of the tube becomes the outside of the roll on the drum. The expressions for strain given in equations (3) and (4) are substituted into equations (1) and (2) so they can be combined for the two conditions of storage on the drum.

When the tube is backward wound, the surface which was the inside of the tube becomes the outside of the roll. Thus, the bending stresses produced by rolling add to those produced by flattening resulting in a transverse stress given by

$$\sigma_x = \frac{E}{1 - \mu^2} \frac{t}{d} + \frac{E}{1 - \mu^2} \frac{t}{D} = \frac{E}{1 - \mu^2} \frac{t}{d} \left(1 + \mu \frac{d}{D}\right) \quad (7)$$

and a longitudinal stress given by

$$\sigma_x = \frac{\mu E}{1 - \mu^2} \frac{t}{d} + \frac{E}{1 - \mu^2} \frac{t}{D} = \frac{E}{1 - \mu^2} \frac{t}{d} \left(\frac{d}{D} + \mu \right) . \quad (8)$$

Reorganizing equation (8) yields the following relation:

$$\frac{d}{t} = \frac{E}{\sigma_x} \frac{\frac{D}{d} + \mu}{(1 - \mu^2) \frac{D}{d}} = \frac{E}{\sigma_x} \frac{1 + \mu \frac{d}{D}}{(1 - \mu^2)} . \quad (9)$$

This is the expression given by F.P.J. Rimrot in reference 3 for determining the allowable tube geometry when σ_x is set equal to the desired allowable stress of the material the tape is made from. A similar expression can be obtained from equation (7) in terms of the longitudinal stress σ_y as given by

$$\frac{d}{t} = \frac{E}{\sigma_y} \frac{\frac{d}{D} + \mu}{(1 - \mu^2)} . \quad (10)$$

In a similar manner, when the tape is forward wound the inside surface of the tape becomes the inside surface of the stored roll. In this case the bending stresses produced by winding subtract from those produced by flattening resulting in the expressions

$$\frac{d}{t} = \frac{E}{\sigma_x} \frac{1 - \mu \frac{d}{D}}{(1 - \mu^2)} \quad (11)$$

and

$$\frac{d}{t} = \frac{E}{\sigma_y} \frac{\frac{d}{D} - \mu}{(1 - \mu^2)} . \quad (12)$$

Typical stress variations using the preceeding equations are shown for backward winding in Figure 2-17 and for forward winding in Figure 2-18. These curves were derived by always setting the critical stress to the allowable value of the material by adjusting the d/t ratio as required.

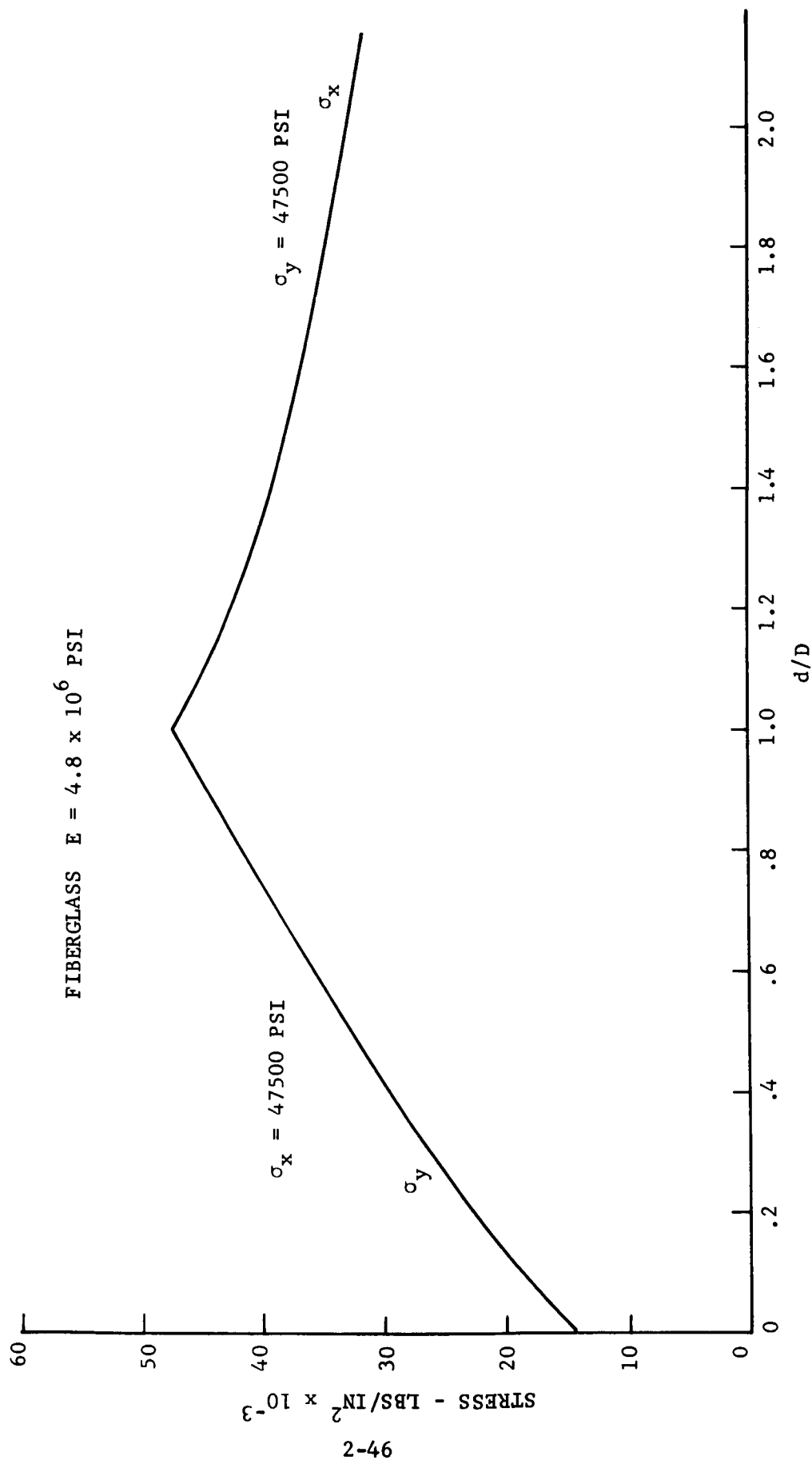


FIGURE 2-17 STRESSES DUE TO BACK WINDING

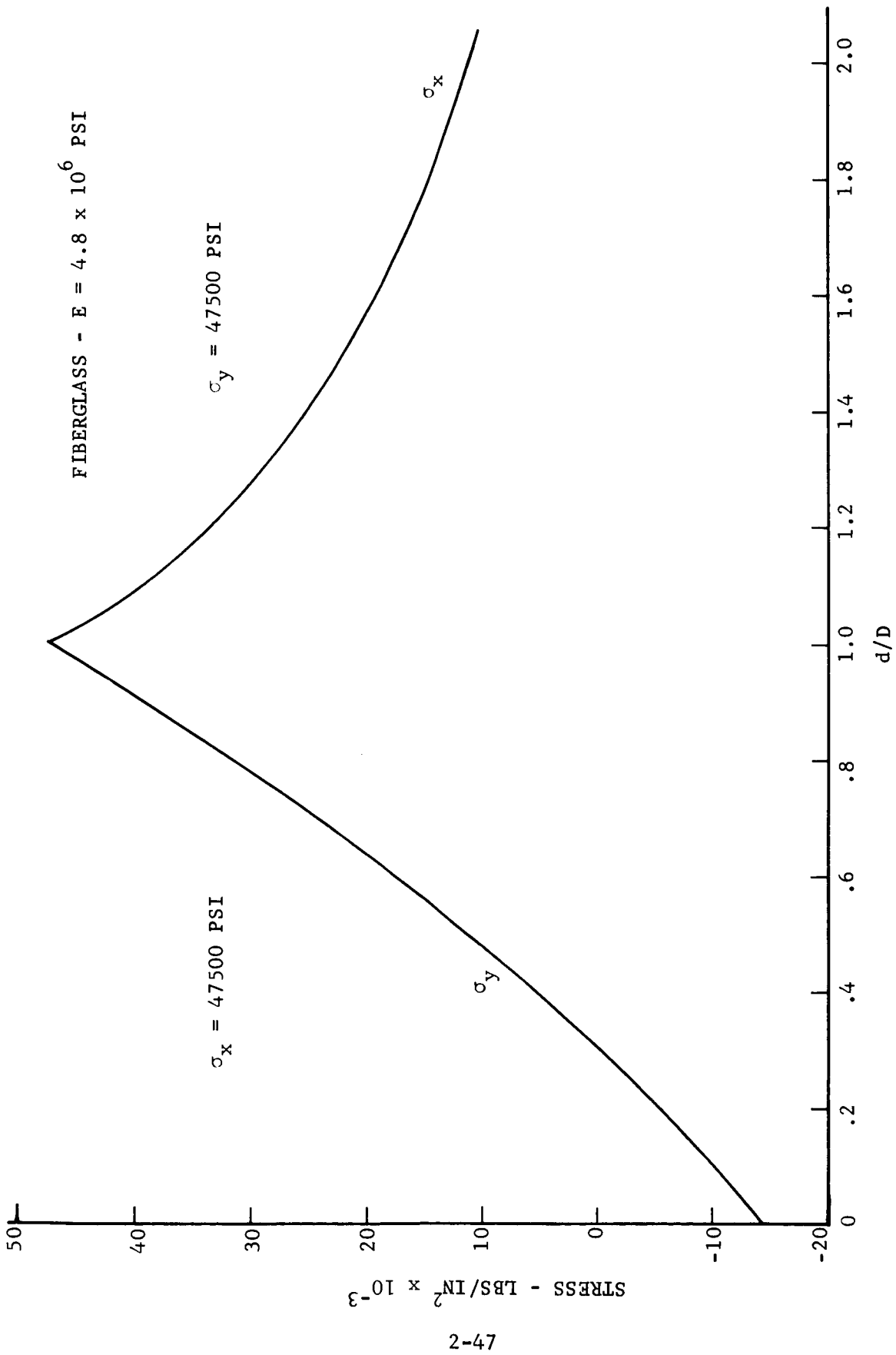


FIGURE 2-18 STRESSES DUE TO FORWARD WINDING

It is interesting to note that in both cases the transverse and longitudinal stresses are equal when the tape is rolled on a drum of the same diameter as the tube. Most commonly the drum diameter is larger than the tape resulting in a d/D ratio between zero and one. Under these conditions the transverse stress due to flattening becomes the critical design stress.

The allowable combinations of geometry such as tube diameter, wall thickness, and drum diameter are in both the backward winding and forward winding cases determined by the transverse flattening stress as long as the drum diameter is greater than the tube diameter, $d/D \leq 1$. For drum diameter less than the tube diameter, $d/D \geq 1$, the longitudinal stress becomes critical. The rate at which the stowage stresses increase are also greater for $d/D > 1$ which indicates that these combinations should be avoided. For backward winding the stowage stress increases as the drum diameter decreases. Thus, the flattening stress determines the allowable tube diameter and wall thickness only for a drum with an infinite radius. As the drum radius decreases, the stowage stress becomes the critical factor which determines the allowable geometry. For a forward wound tube, the stowage stresses decrease as the drum diameter is reduced until a minimum is reached when the drum and tube are equal in diameter. As the drum is decreased in size below the tube diameter, the stowage stresses increase until they again equal the flattening stress when the drum reaches a size determined by Poisson's ratio, $d/D = 1 + \mu$. These relationships are graphically shown in Figure 2-19 for a typical material. The fact that the stowage stresses are less than the flattening stress cannot be utilized to reduce the d/t ratio allowable since they are arrived at sequentially. The flattening stress must always be endured before the stowage stresses are imposed.

The primary benefit of forward winding is that a minimum size drum can be used up to the critical value determined by the Poisson ratio of the material.

In reference 3, F.P.J. Rimrot indicated that an optimum drum to tape diameter ratio occurred at a value of 3.33 which would yield a minimum d/t ratio. By examining the stress equation (12), it is seen that this is the point of minimum stress for σ_y , the longitudinal stowage stress. This occurs when d/D is equal to Poisson's ratio μ which reduces this stress to zero. This is the point at which the critical stress produced by winding on a drum equals that produced by flattening the tube. As was previously pointed out, these stresses are experienced sequentially so that if a design were to utilize this minimum d/t ratio yielding would occur somewhere between flattening and winding the tape on the drum. Subsequent unfurling of the tape would result in a tube with a larger diameter. Thus, the d/t ratio would adjust itself to the proper value. The allowable combination of d/t and D/d ratio were plotted and compared to the data in reference 3. This is shown in Figure 2-20. Perfect agreement is achieved for backward winding but not for forward winding. No explanation for the differences shown for forward winding could be discovered.

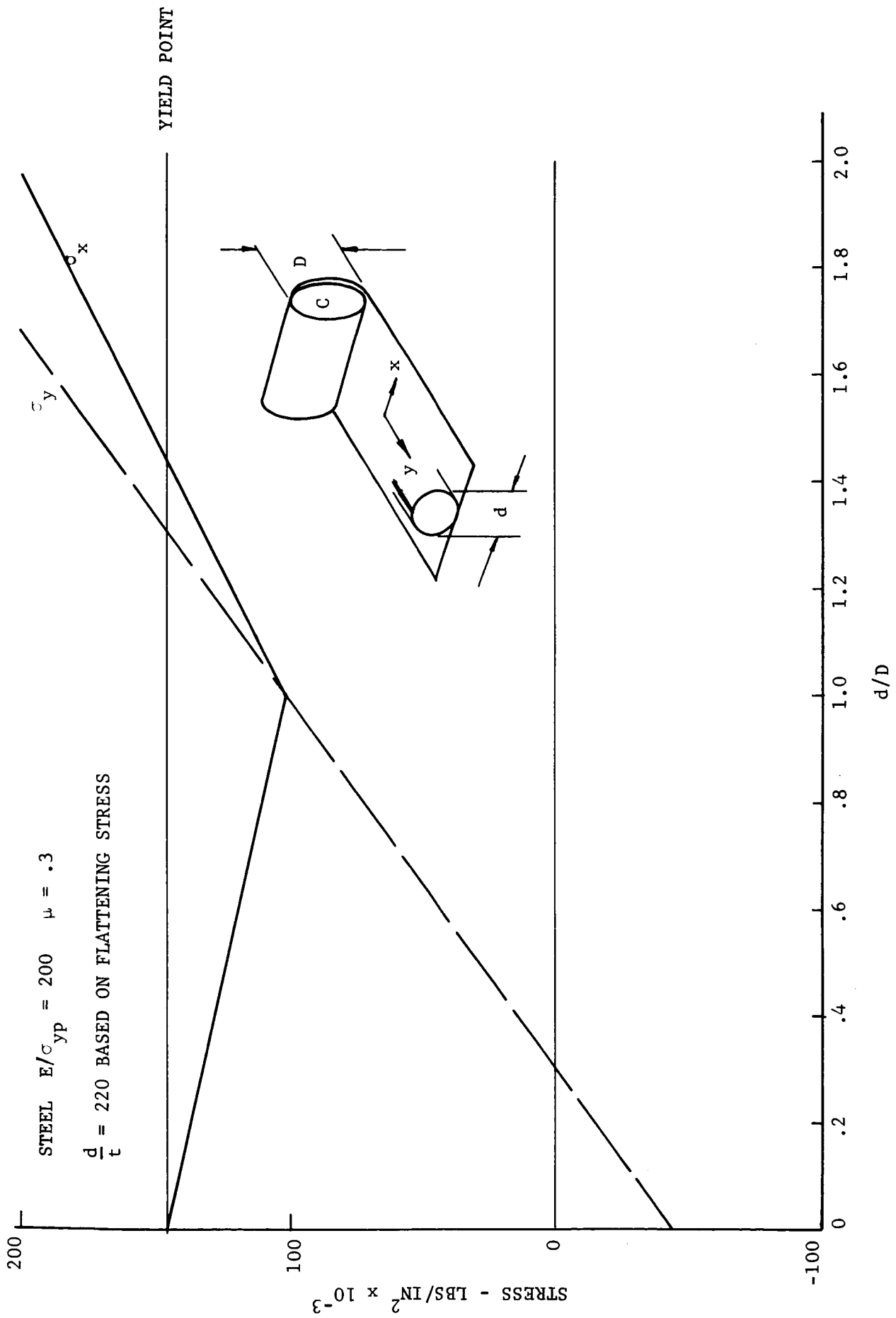


FIGURE 2-19 STRESS VARIATION WITH DRUM SIZE FORWARD WINDING

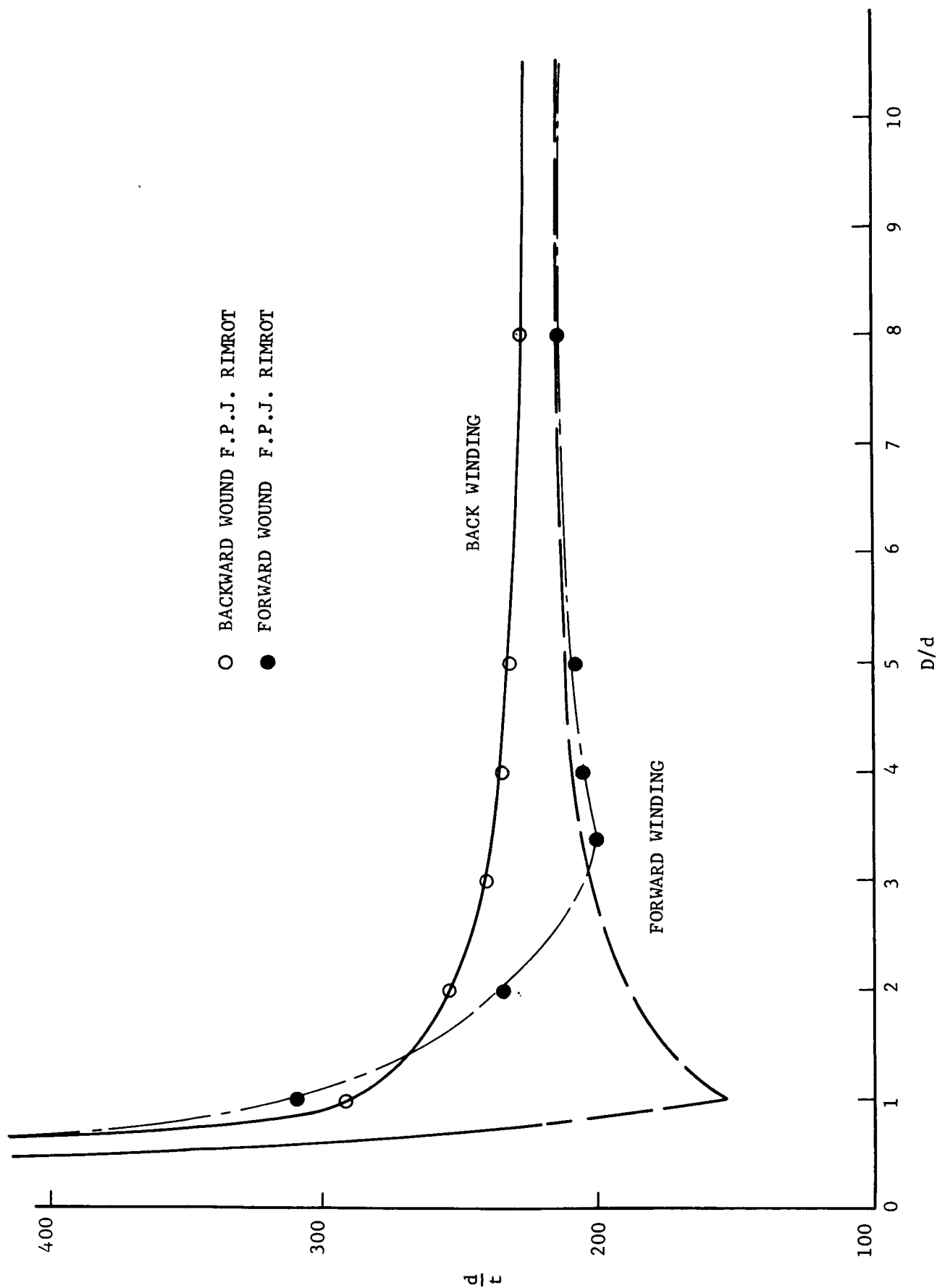


FIGURE 2-20 COMPARISON OF DATA PRESENTED BY DEHAVILLAND IN DEC. 9, 1965 MACHINE DESIGN

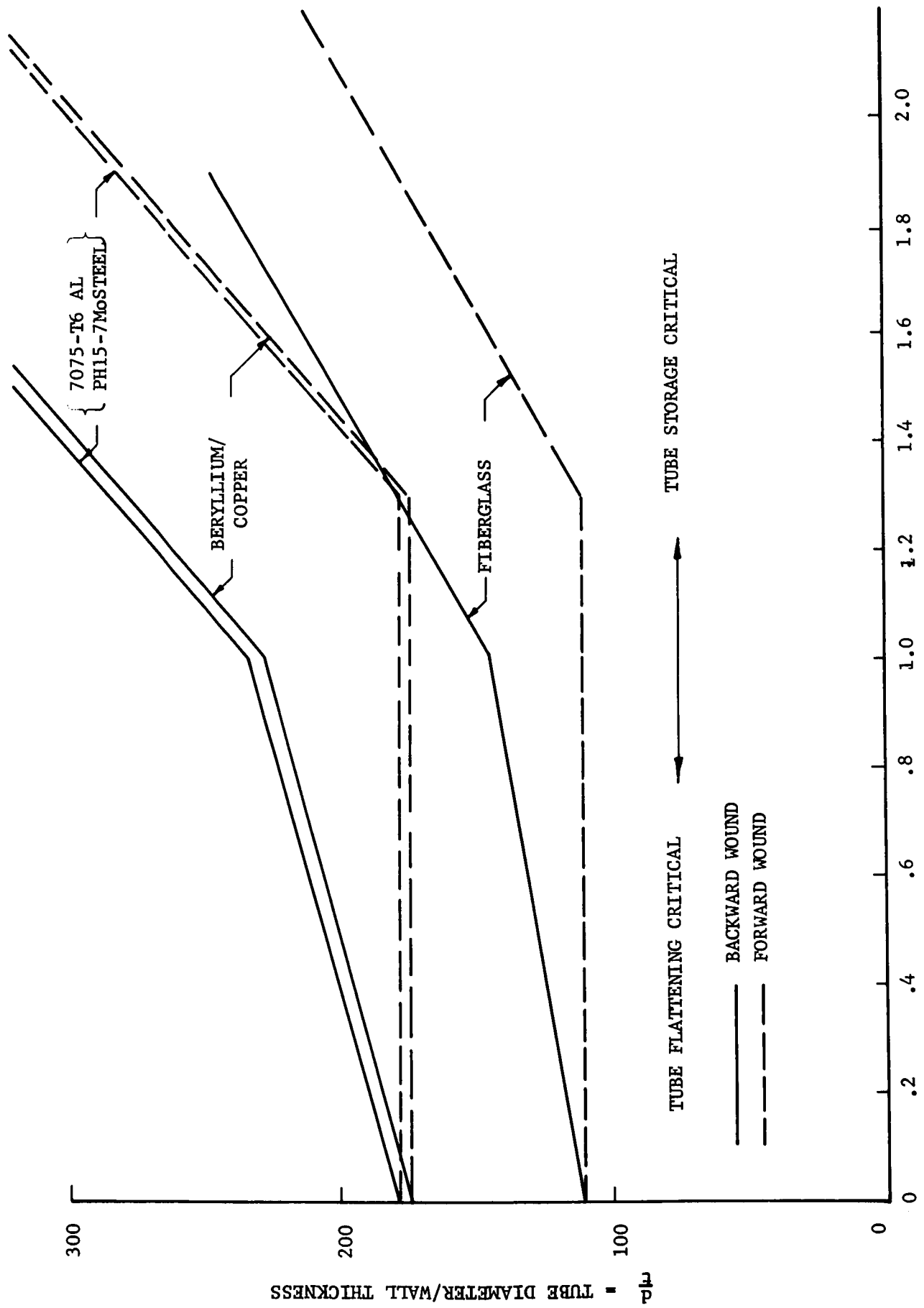
Figure 2-21 summarizes the allowable geometric combinations for the three basic materials investigated in this study. It is seen that fiberglass produces the lowest allowable d/t ratio for a given d/D ratio. This means that the thickest walled tube for a given tube and drum diameter can be achieved with the fiberglass. This is desirable from the standpoint of achieving resistance to buckling. As was shown earlier in the parametric analyses, fiberglass also will result in a minimum weight boom for a given deployed range. Whether or not this apparent advantage of fiberglass can be realized will depend on developing fabricating techniques and testing it as a furlable tape.

A preliminary design concept utilizing the furlable tape concept was generated and is shown in Figure 2-22. This boom concept was generated for the spherical abrading sieve sampling concept 8C included in Appendix B. The operational sequence of the furlable boom is given by Table 2-X. It is assumed that only a command is required to initiate action and only one motor is used to drive all motions. It is also assumed that the boom starts in a stowed position from which it is required to rotate in a horizontal plane and elevate in a vertical plane before deployment begins.

TABLE 2-X
OPERATIONAL SEQUENCE

1. Initiation command.
2. Rotate boom in horizontal plane to align it along a radial line outward from the payload center.
3. Elevate to 45° from horizontal. This may occur simultaneously with step 2.
4. Extend boom to maximum length.
5. Reduce elevation until contact with surface is achieved.
6. Initiate sample collection and radial traverse of $\pm 30^{\circ}$ as well as retraction of the boom.
7. Elevate boom to 45° and complete retraction when geometrical limits or time limits dictate termination of sampling.

The operational sequence outlined in Table 2-X suggests several additional mechanizations. The initial elevation and rotation may be spring driven because it is a single event. The $\pm 30^{\circ}$ radial sweep can be continuously acting. A slip clutch can be incorporated to protect the boom from failure by lodging against obstacles. Limit stops will be necessary to keep the boom within the $\pm 30^{\circ}$ excursions after slipping. By combining extension and upward elevation, retraction and downward elevation can be achieved with the same gear train without commands or actuating clutches.



d/D - TUBE DIAMETER/DRUM DIAMETER

FIGURE 2-21 ALLOWABLE FURLABLE TUBE GEOMETRY

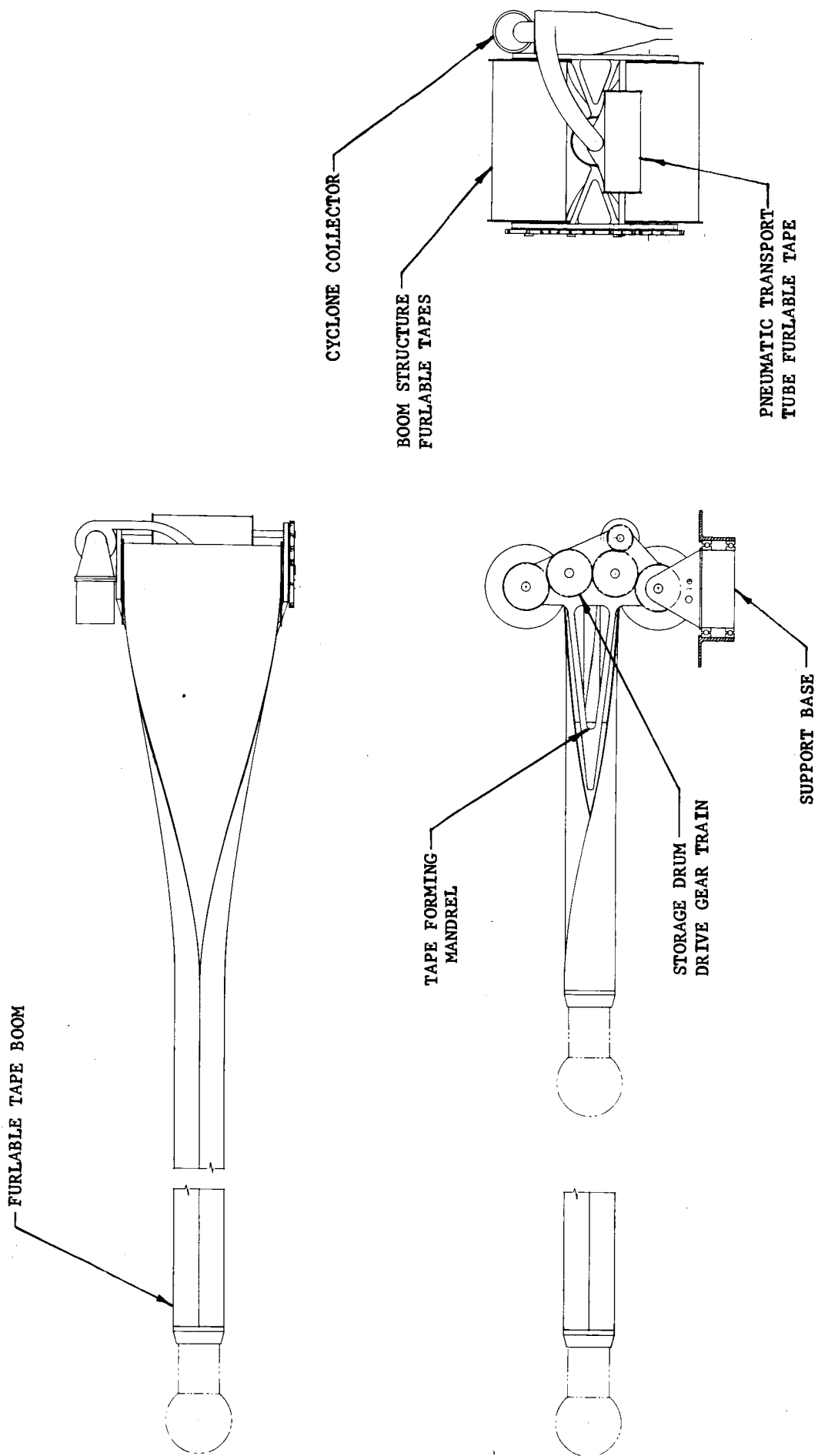


FIGURE 2-22 FURLABLE BOOM CONCEPT

A slip clutch on the elevation mechanism is required to provide a constant force between the end of the boom and the surface. Under the assumption that deployment to a distance of 20 feet requires one minute, then the rotational speed of the drum is $N - 240/6 = 40 \text{ rev/min}$.

In order to regain some degree of torsional strength, this design proposes to use to furlable tapes that fold around one another as shown in Figure 2-23.

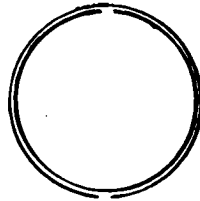


FIGURE 2-23. CROSS-SECTION THROUGH DOUBLE FURLABLE TAPE

This configuration will make the shear center and geometric center of the tube coincide. This configuration depends on the friction forces between the two tubes to restore torsional strength. Techniques such as embossing the surfaces so that some mechanical interlocking occurs could be used to achieve higher torsional strength. For this design a tube diameter of 1.5 inches with a wall thickness for each tape of .010 inches was used with a D/d ratio equal to one. For this geometry a flattened tape width of 4.75 inches is required. To estimate the loaded drum size, a boom length of 20 feet was assumed. This added .5 inches to the drum radius resulting in an overall drum size of 2.5 inches diameter and 4.75 inches long. These are rather large drums, but the specific packaging and operational requirements will determine whether the size is objectionable. It is pointed out in Figure 2-22 that a third smaller furlable tape is mounted concentrically with the boom tapes. The intent here was to explore the mechanization of providing a smaller diameter pneumatic transport tube if such were found to be desirable. In order to provide a reasonably air tight tube, this tape is wide enough so that it can fold around itself as shown in Figure 2-24.

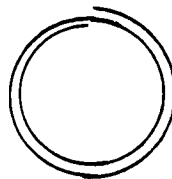


FIGURE 2-24. CROSS-SECTION OF PNEUMATIC TRANSPORT TUBE

A furlable tape of this configuration has been manufactured by DeHavilland of Canada. A tube diameter of .5 inches with .001 wall thickness on a one inch diameter drum is assumed. This results in a tape width of 3.8 inches which is compatible with the larger boom tapes. As shown in Figure 2-22, all three tape drums are driven through a spur gear train. Since the large tapes are thicker and are wound on a larger drum the deployment rate for these tapes is different than for the smaller tape. The feed rate for the larger tape starts at 7.85 inches per revolution with a full drum and terminates with a feed rate 4.72 inches per revolution for an empty drum. The corresponding feed rate for the small tape is 3.19 inches per revolution and 3.14 inches per revolution. A two to one gear ratio can be used to drive the small drum. This will make the average values of feed rate correspond; however, they cannot be made to correspond exactly over the entire range of deployment. This is shown in Figure 2-25. It is seen that the larger tape deploys more rapidly initially and more slowly at the termination of deployment. This can be compensated for by spring mounting the small tape drum on its shaft. This will allow relative rotation of the tape drum to its shaft thereby allowing the smaller tape feed rate to adjust to that of the large tape. If an initial preload is provided in this spring mounting, the small tape will always be in tension which would be desirable from the standpoint of stabilizing it and preventing it from buckling. The tension will build up to a maximum at the midpoint of deployment and then reduce to a minimum at the end of deployment. This is shown schematically in Figure 2-26.

It is apparent that if no development problems occur in fabricating the tapes themselves, this concept could be much simpler than a pneumatically deployed telescoping boom and hence more reliable.

To terminate the study effort on deployable booms, an attempt was made to find out if deployable booms suitable for deploying a soil sampler were currently available from an outside source. It was for this reason that on 1 June 1966, W. H. Bachle and D. H. Garber made a trip to San Diego to investigate the application of furlable booms developed by General Dynamics/Convair and Ryan Aeronautical Company. The booms being fabricated at General Dynamics/Convair were developed for use as satellite gravity gradient stabilization devices. Their minimal structural strength requirements rendered them unfit for general purpose early Voyager-class sample handling system utilization. The furlable booms being developed at Ryan are directly applicable to the sampler deployment problem. Ryan's boom elastically returns to shape because it utilizes a closed cross-section with good rigidity in both bending and torsion. The tape is fabricated from two thin strips of metal seam-welded at both edges and preformed to close into tubular shape as it unwinds from a drum. Compared to a de Havilland furlable tube, this boom will need a larger sized drum and more retracting power. Ryan engineers estimated that approximately

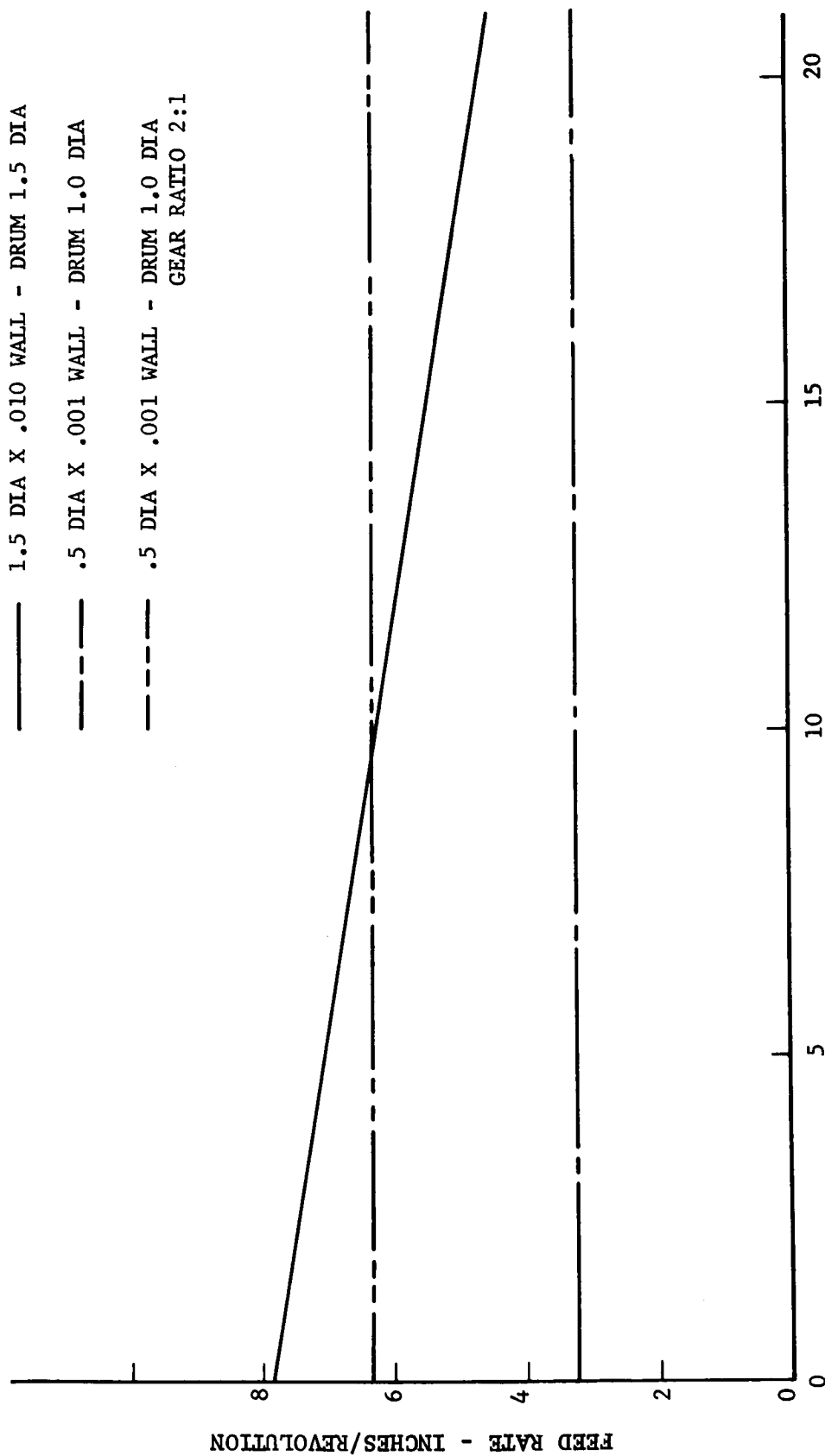


FIGURE 2-25 DEPLOYMENT RATE CHARACTERISTICS FOR FURLABLE BOOM

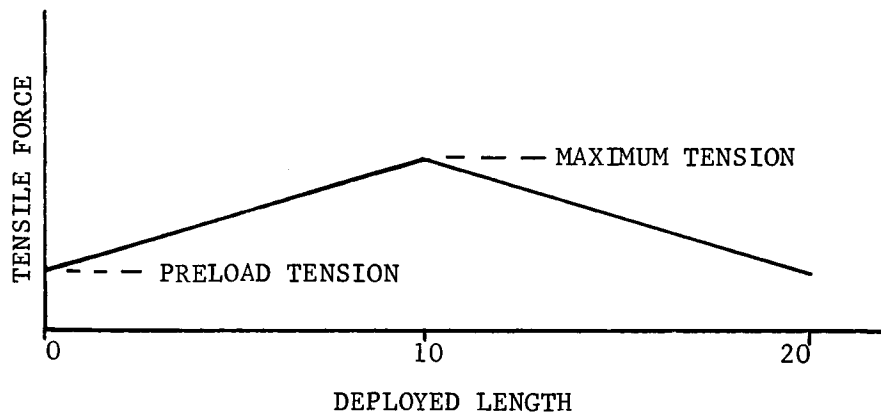


FIGURE 2-26. SCHEMATIC DIAGRAM OF PNEUMATIC TAPE TENSILE FORCE VARIATION

four months would be required to design and fabricate a furlable boom to meet the sample deployment subsystem requirements. This schedule was not compatible with ABL schedules. Since Ryan does not manufacture the tape as a commercial item, it did not appear practicable to use it at that time for the ABL sampling program.

2.3.2 PNEUMATIC MOTORS AND COMPRESSORS

Early in the conceptual design studies it appeared that information on the tradeoffs between weight, volume and efficiency factors for electrical motors and pneumatic motors powered by stored compressed gas or by an atmospheric compressor would be desirable. The application of electrical motors within the stipulated power constraint would require high rpm low torque motors with gear reduction trains. The power output, efficiency and reliability of these systems for extra-terrestrial use are well documented. A multiple cylinder pneumatic motor can produce high torque at very low rpm. In this case a gear reduction train may be completely eliminated and the efficiency and reliability of the system markedly increased. For sample handling system purposes, these factors are of utmost importance. Certain complexities are introduced into a pneumatic system, however. These have to do with the positioning of compressed gas tanks and for the pneumatic tubing leading from centrally located tanks or compressor station.

A minimum effort study was undertaken to define the geometry, power output, and flow requirements of multiple cylinder air piston motors. The application of this type of motor as a compressor was also investigated. From these data, a comparison between electrical and pneumatic motors in terms of power supply relative to weight can be derived.

In designing an air piston motor for Martian sample handling system use, the following assumptions are made:

- (1) A combined cycle, as shown in Figure 2-27, is used to estimate work.
- (2) The crank arm r is equal to the cylinder radius $d/2$.
- (3) Nitrogen is used as a working fluid.
- (4) Clearance volume is determined for X_0 equal to five percent of the stroke ($.1 r$).
- (5) A minimum length connecting rod is used for compactness, $1 - 3 r$.

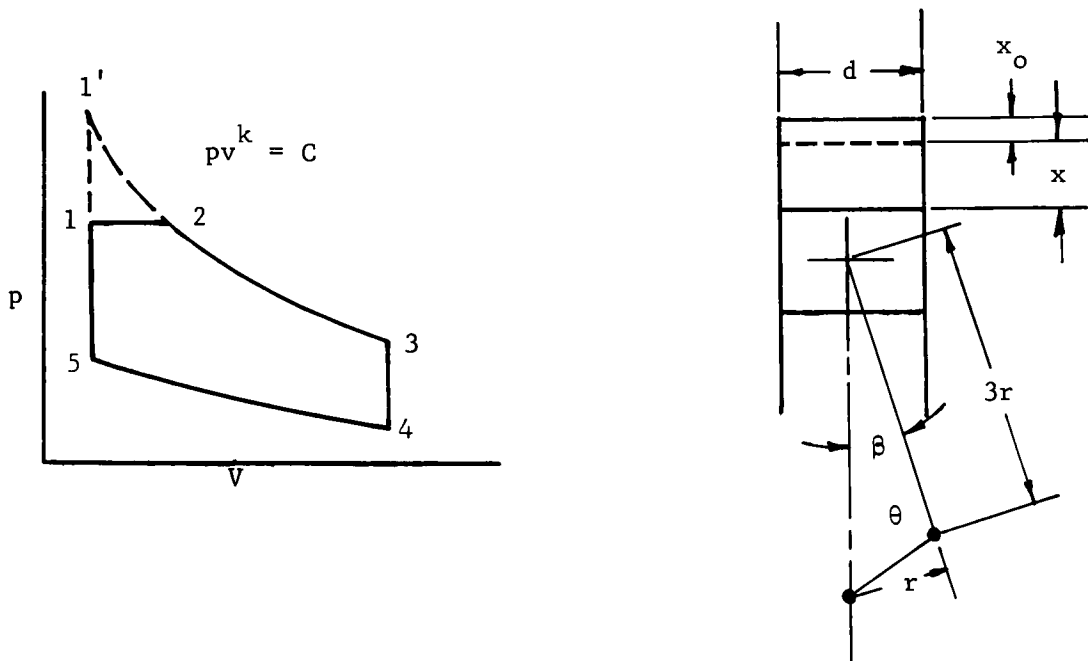


FIGURE 2-27. OPERATIONAL CYCLE AND PISTON, ROD AND CRANKSHAFT PARAMETERS FOR AN AIR PISTON MOTOR

The following events occur at various points in the cycle.

- (1) Intake open. Power stroke begins, intake remains open.
- (2) Intake closes, power stroke continues.
- (3) Power stroke ends, exhaust opens.
- (4) Exhaust closes, return stroke begins.
- (5) Return stroke ends, intake opens.

Note in Figure 2-27, the cycle shown to 1' closes the intake at 1' rather than 2 which will provide a more complete expansion of the gas. This would be the case where the intake valve opens and closes within a few degrees of crank rotation. The cycle shown is based on the assumption that with the use of multiple cylinders, a high torque low rpm motor can be made to produce high initial power and, throughout the work cycle, to produce power smoothly. Thus, the use of gear reduction trains may be eliminated resulting in higher reliability and efficiency.

The following relations hold for the assumed motor geometry defined in Figure 2-27. The connecting rod and crankshaft angles are related by

$$\sin \beta = \frac{\sin \theta}{3} \quad (13)$$

and the stroke is given by

$$x = r(1 - \cos \theta) + 3r(1 - \cos \beta) . \quad (14)$$

The displacement volume is given by

$$V = A(x + x_0) = \pi r^3 \left[(1.1 - \cos \theta) + 3(1 - \cos \beta) \right] \quad (15)$$

The work performed in a cycle is the area contained within the cycle diagram which is the sum of the work performed in a constant pressure expansion from 1 to 2, an isentropic expansion from 2 to 3, and the isentropic compression from 4 to 5 given by

$$W_{\text{net}} = p_1(V_2 - V_1) + \left[\frac{p_3 V_3 - p_2 V_2}{1 - k} \right] - \left[\frac{p_5 V_5 - p_4 V_4}{1 - k} \right] \quad (16)$$

The torque produced throughout the power stroke is given by

$$T = Ar \sin \theta = \pi p r^3 \sin \theta . \quad (17)$$

To estimate the power output, torque characteristics, and gas flow requirements the following motor configuration was assumed.

- (1) The motor has six cylinders.
- (2) The cylinder diameter is .25 inches.
- (3) The pressure of the gas supply is 1000 psi.

The torque characteristics of this motor were investigated first. Three values for the amount of crankshaft rotation which would occur while the intake valve was open were also assumed as follows.

- (1) An instantaneous opening and closing at top dead center. This is the theoretical cycle which goes to 1' in Figure 2-27.
- (2) The intake of one cylinder remains open for a crankshaft rotation of 60 degrees or until the next cylinder reaches top dead center.
- (3) The intake of one cylinder remains open for a crankshaft rotation of 65 degrees. This provides a 5 degree overlap in which the intake to two cylinders are simultaneously open.

The torque characteristics for these three conditions are shown in Figures 2-28, 2-29, and 2-30, respectively, as a function of crankshaft rotation angle. The torque contributions of each cylinder are shown below the upper curve which is the sum of these contributions. It is readily apparent that an intake valve opening of 60 degrees crankshaft rotation produces the most uniform torque output. The 5 degree overlap which it was thought to produce a more uniform torque produced sharp peaks on an otherwise fairly constant torque curve. In either case the torque appears uniform enough so that the motor could be driven at very low speeds. It is interesting to note that the absolute value or magnitude of the torque for the instantaneous valve opening was nearly an order of magnitude less than for the other two cases. The first cycle is probably thermodynamically more efficient than the latter two; however, thermodynamic efficiency is probably not a primary criteria for small motors with unique operating characteristics. A point of interest is that even though the assumed dimensions were extremely small, the latter two motors can produce an absolute level of torque of 7 inch pounds or on the

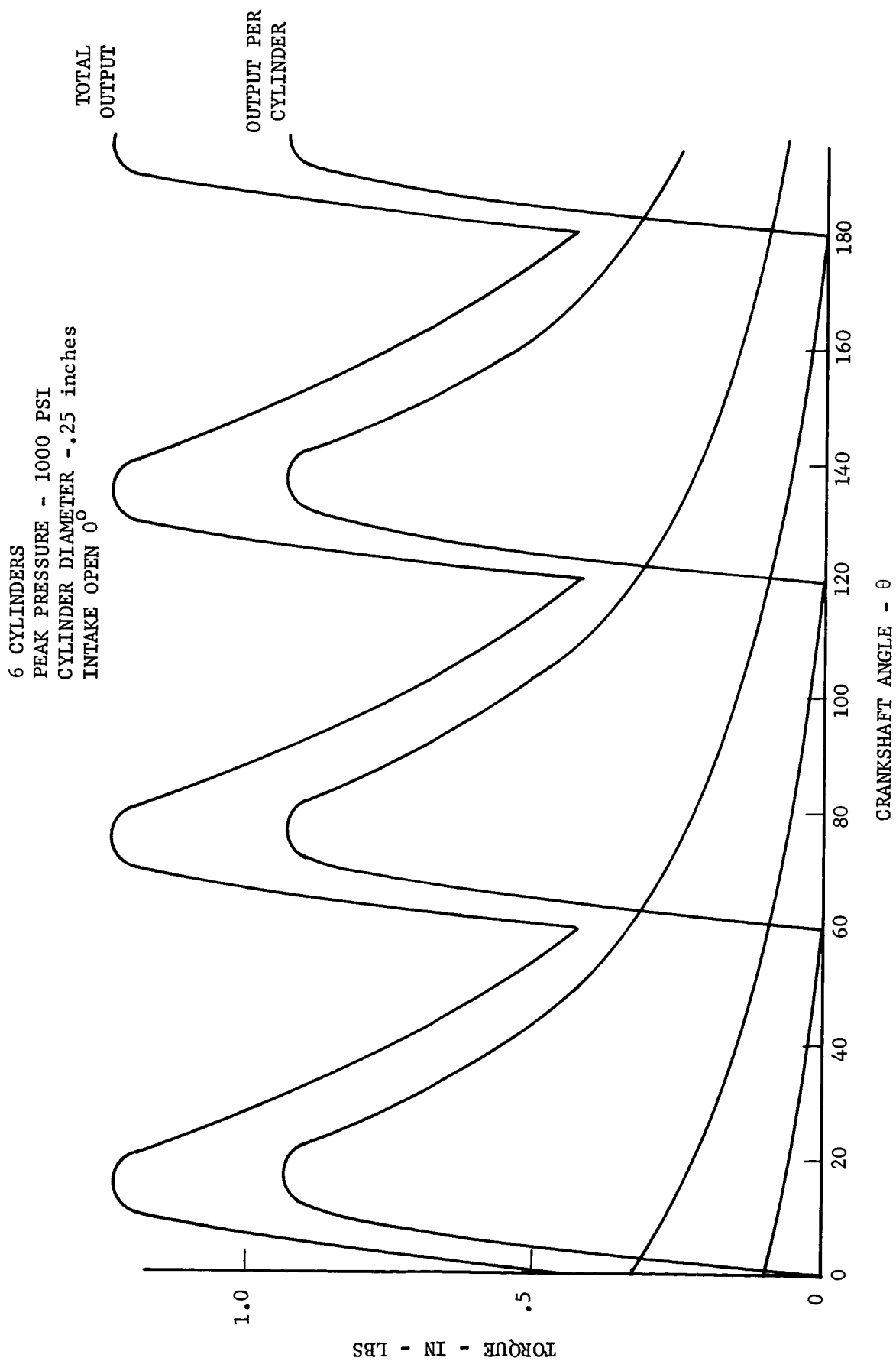


FIGURE 2-28 TORQUE CHARACTERISTICS OF PNEUMATIC MOTOR NO. 1

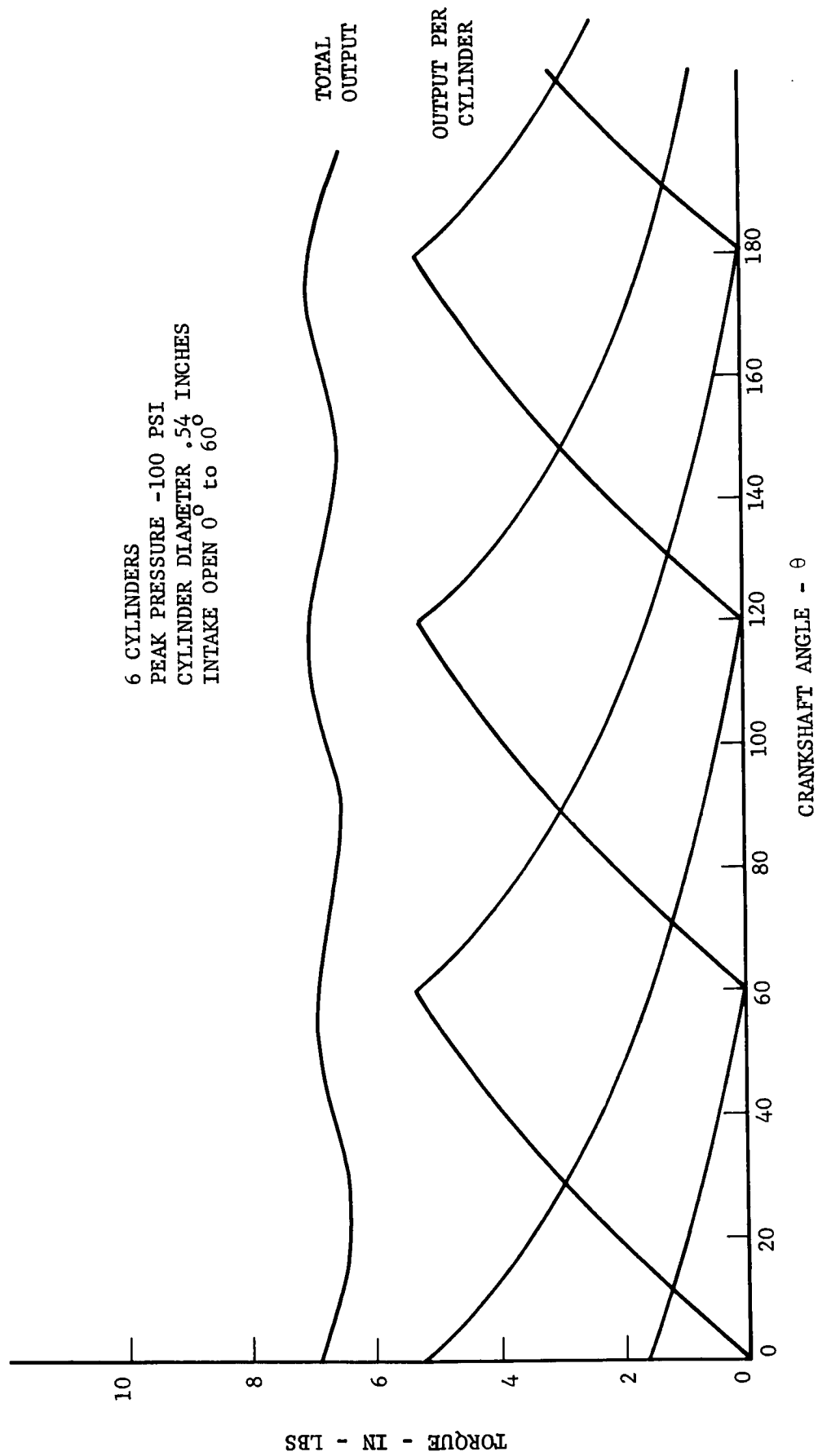


FIGURE 2-29 TORQUE CHARACTERISTICS OF PNEUMATIC MOTOR NO. 2

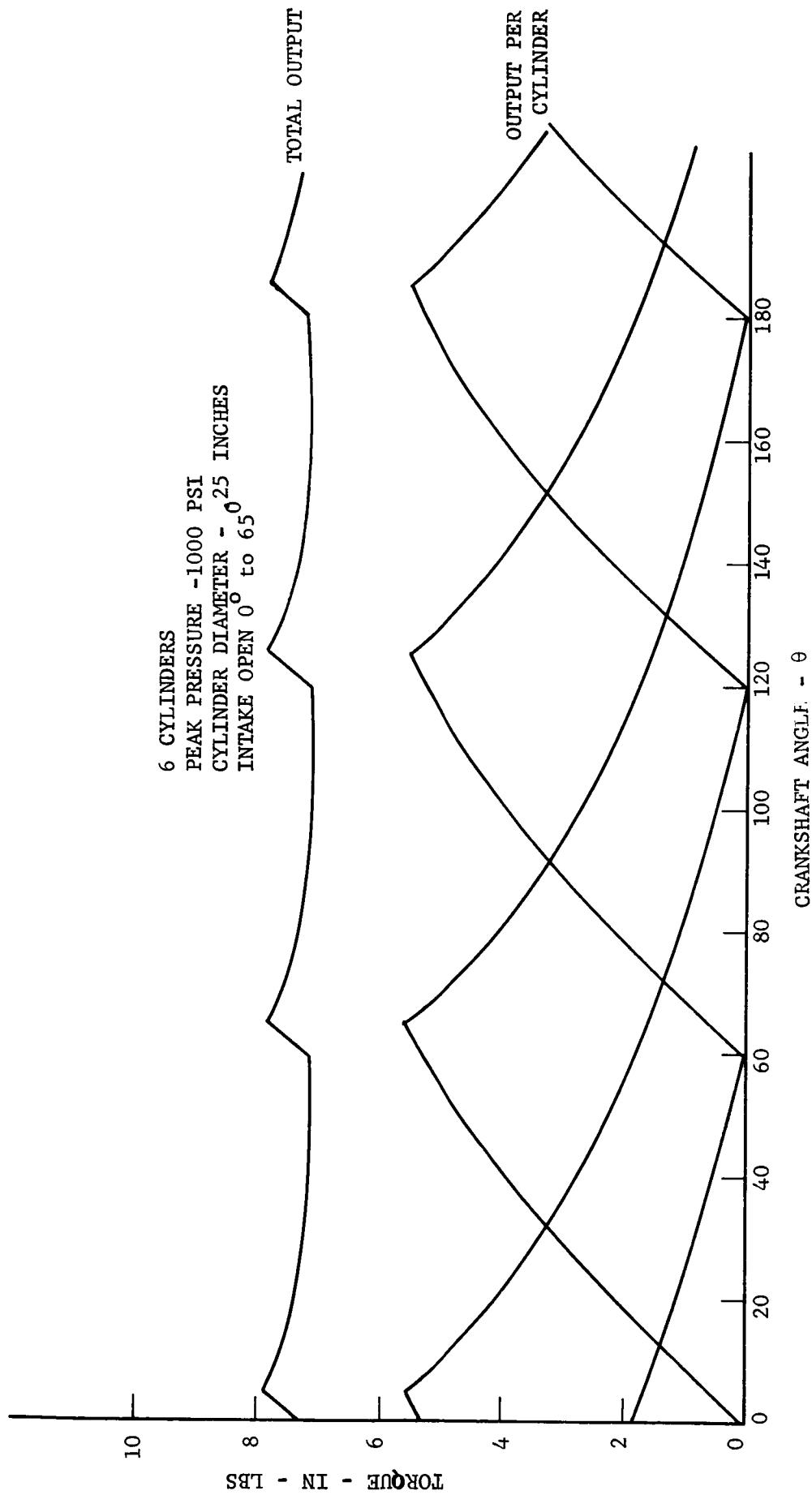


FIGURE 2-30 TORQUE CHARACTERISTICS OF PNEUMATIC MOTOR NO. 3

order of 110 inch ounces at the shaft independently of motor speed. This is comparable to the output of a high speed electric gear motor with a gear reduction ratio in the range of 100 to 1. The electric motor torque is developed at a fixed output shaft speed depending on the motor rpm and the gear reduction ratio. Thus, the pneumatic motor offers the potential elimination of the gearbox and variable speed drive by adjusting the flow rate of the gas supply.

For the assumed gas supply pressure, a residual exhaust pressure of 226 psi is achieved which is still fairly high. If the main gas supply pressure is reduced to 100 psi, the residual exhaust pressure drops to 22.6 psi which is much more reasonable. In order to operate at a reduced gas supply pressure, the physical size of the motor must be increased. To achieve the same torque a larger diameter cylinder must be used. If the same geometric proportions are retained, the crankshaft throw also increases. To arrive at a scaling factor, it is noted that the torque is proportional to the gas supply pressure and the crank arm as given in equation (17). Thus, the following relation can be written

$$p_1 r_1^3 = p_2 r_2^3 \quad (18)$$

from which the cylinder radius and crank arm dimension can be determined by

$$r_2 = r_1 \sqrt[3]{\frac{p_1}{p_2}} \quad (19)$$

For a gas supply pressure of 100 psi, the diameter of the cylinder must be increased by the cube root of ten or 2.16 times. This is slightly more than double the size required for a 1000 psi gas supply. Figure 2-31 graphically displays the relationship between motor speed, output power, and gas supply flow rate. Horsepower up to a tenth can be achieved at moderate speeds for fairly low gas supply flow rates.

A comparison was made for the system weight variation of a battery powered electrical system and a nitrogen gas supplied pneumatic system. Silver-zinc batteries were used as a basis in the electrical system. The gas supply plus its storage tank weights were used as a basis in the pneumatic system. These weight comparisons are shown in Figure 2-32. It is noted that the battery powered system weight can vary over a large range depending on the discharge rate of the batteries. For soil sampling applications, the power demands and hence discharge rate can become appreciable. Based on the results shown in Figure 2-32, it is seen that the battery powered electrical system weight is less for a given total energy expenditure. It should be noted that motor weights were not included in this analysis. Since the pneumatic motor does not require the use of heavy materials such as magnets and copper wire used in motors, the motor should

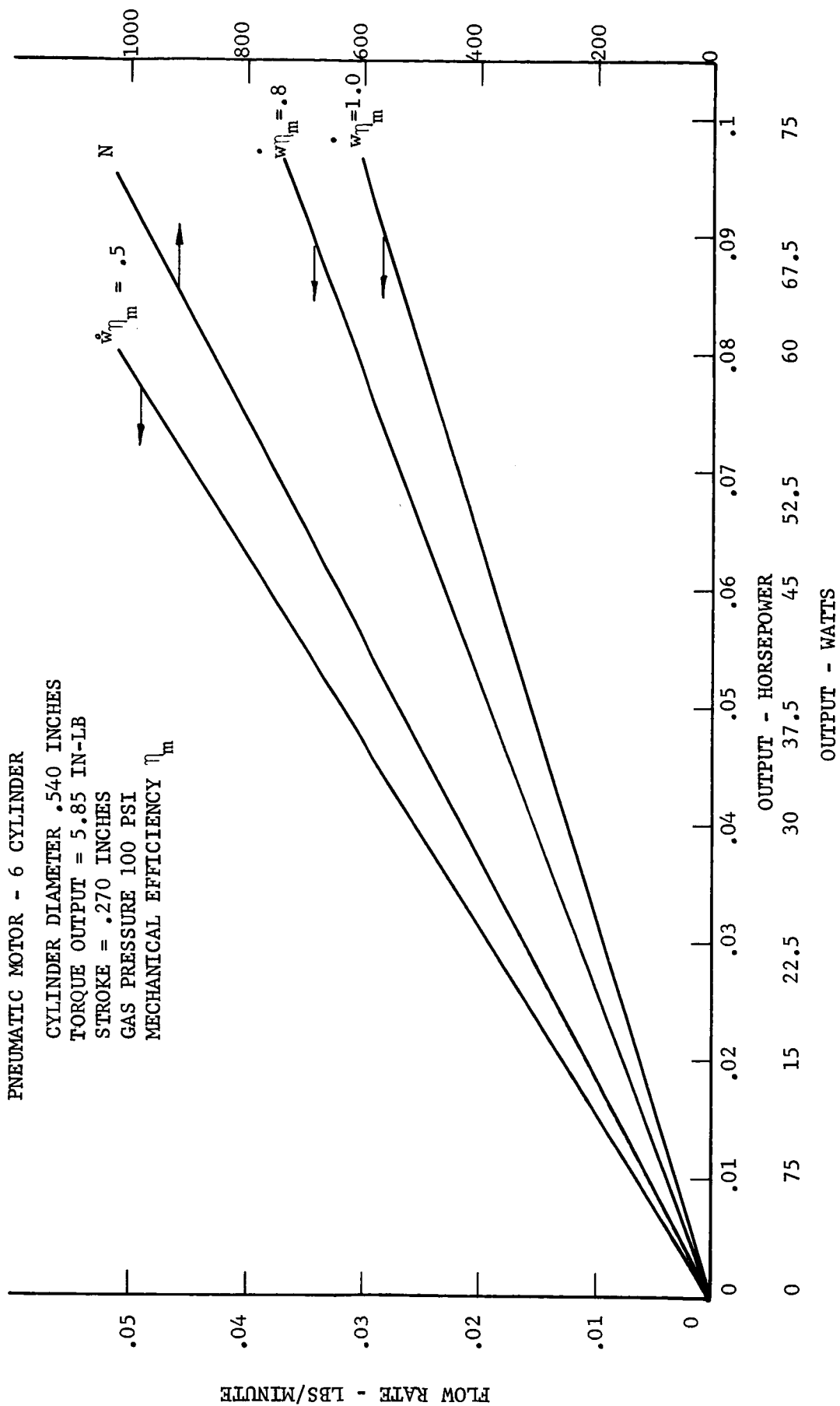


FIGURE 2-31 FLOW RATE & HORSEPOWER CHARACTERISTICS

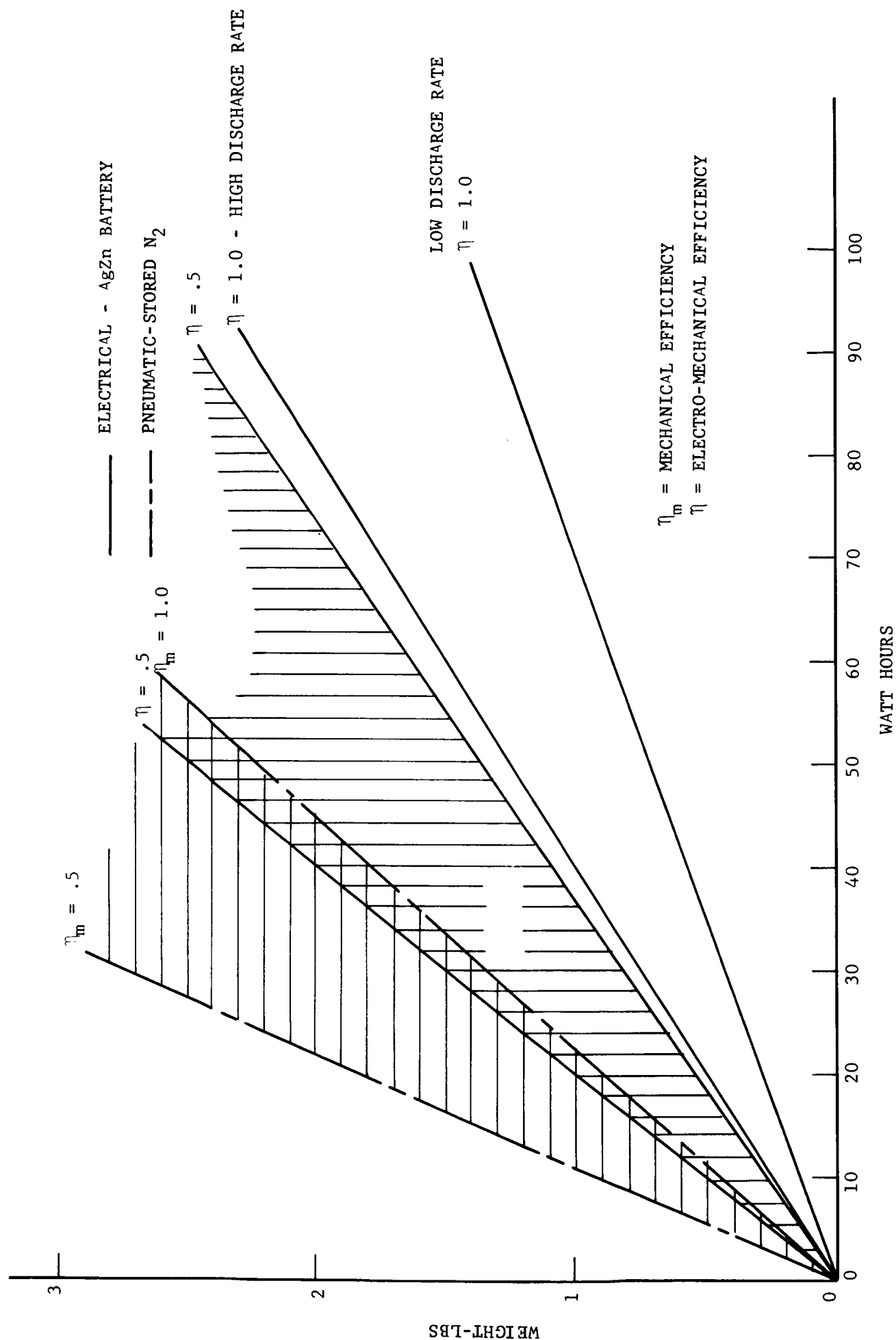


FIGURE 2-32 COMPARISON OF POWER SUPPLY SYSTEM WEIGHT VARIATION

be considerably lighter. This will make the pneumatic system compare more favorably with the electrical system, particularly when the total energy demand is low but the power and torque requirements are high.

If power is applied to a motor, it can be made to work as a compressor. As a matter of interest, the possible use of this motor as a compressor was investigated. The work cycle for such a compressor, using the same valving system as the motor used, is shown in Figure 2-33.

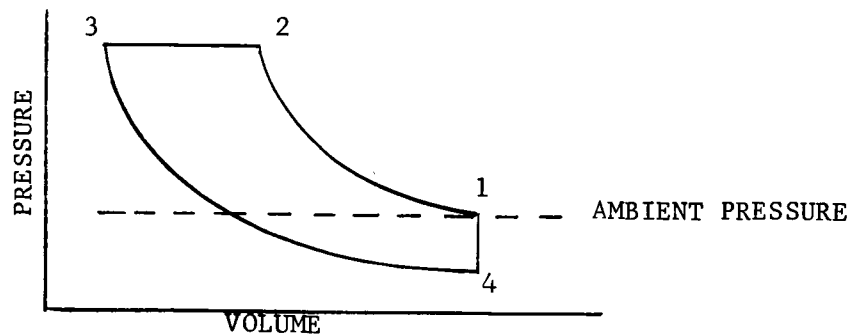


FIGURE 2-33. WORK CYCLE FOR PNEUMATIC MOTOR RUN AS A COMPRESSOR

The steps in this work cycle are as follows.

- (1) The intake valve opens and admits atmospheric gas into the cylinder at ambient pressure.
- (2) The air is compressed isentropically from 1 to 2 in the work cycle.
- (3) At point 2, the exhaust valve opens and the compressed gas is discharged at constant pressure until the piston reaches top dead center at point 3.
- (4) The exhaust valve closes and the remaining gases expand isentropically to point 4 which is below ambient pressure. The cylinder is recharged and the cycle is repeated.

Since it is anticipated that free air flow rates covering the range of values shown in Figure 2-34 will be required for pneumatic transport systems, the cylinder size required in this type of compressor was determined for several rotational speeds in a 10 millibar atmosphere. The variation of cylinder size is shown in Figure 2-35 for a six cylinder compressor. From these curves, it is seen that the fairly large cylinder diameters are required to produce appreciable flow rates in a Martian environment.

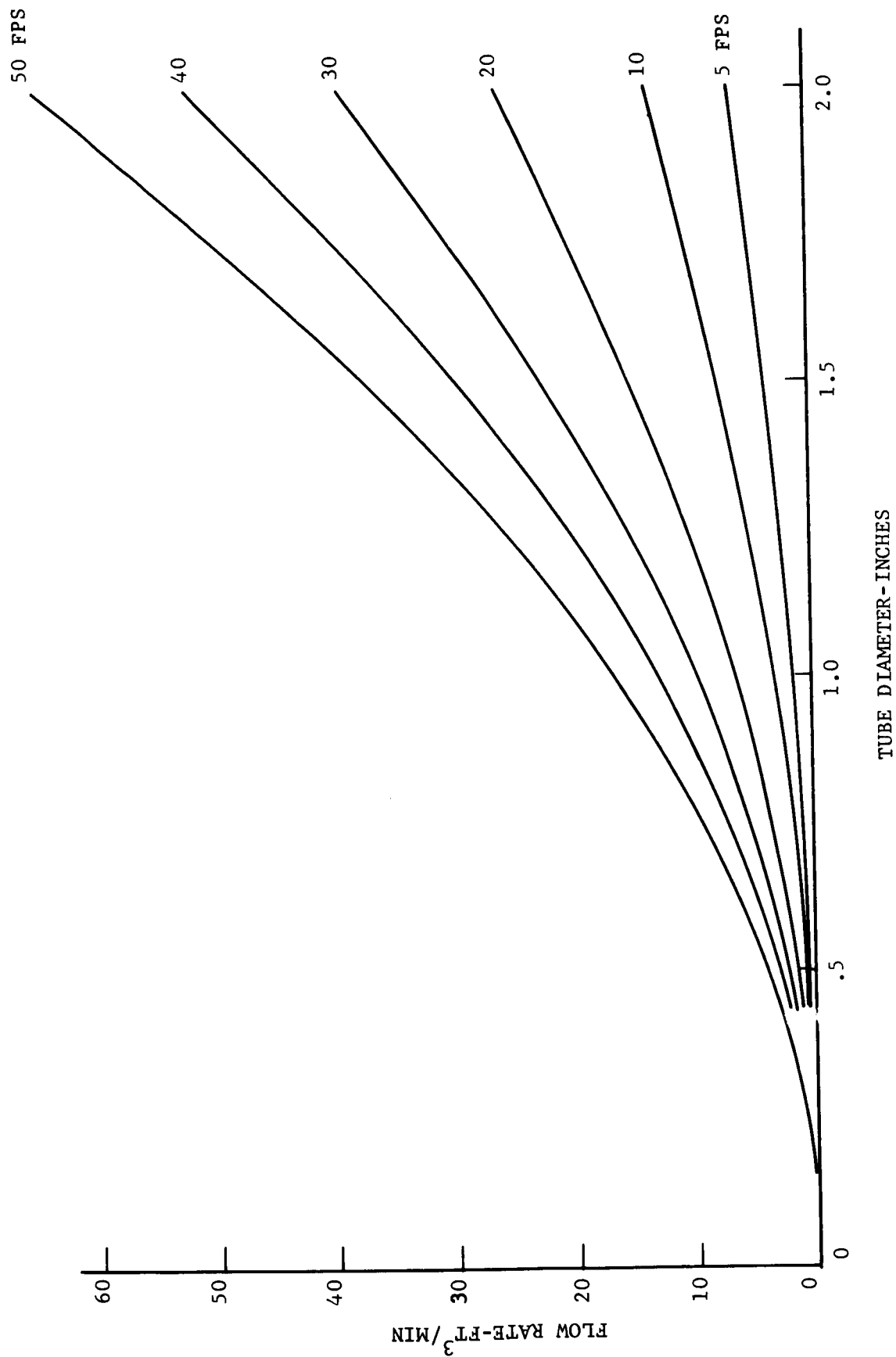


FIGURE 2-34 FREE AIR FLOW RATE VARIATION WITH TUBE DIAMETER

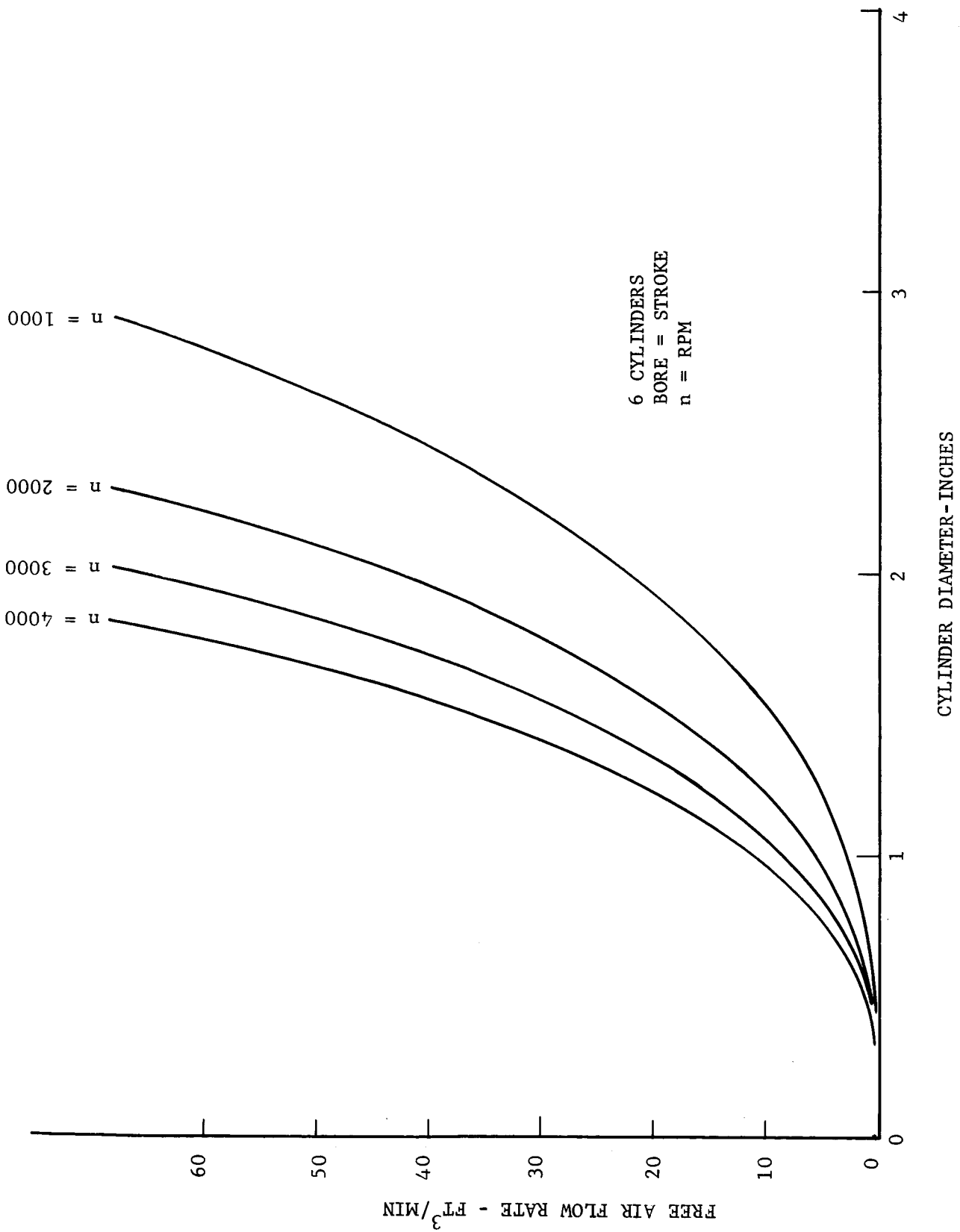


FIGURE 2-35 COMPRESSOR CYLINDER DIAMETER REQUIRED

The valving for the compressor assumed in the calculations above is not that conventionally used in a piston compressor. The work cycle and valve-piston configuration for a conventional compressor is shown in Figure 2-36.

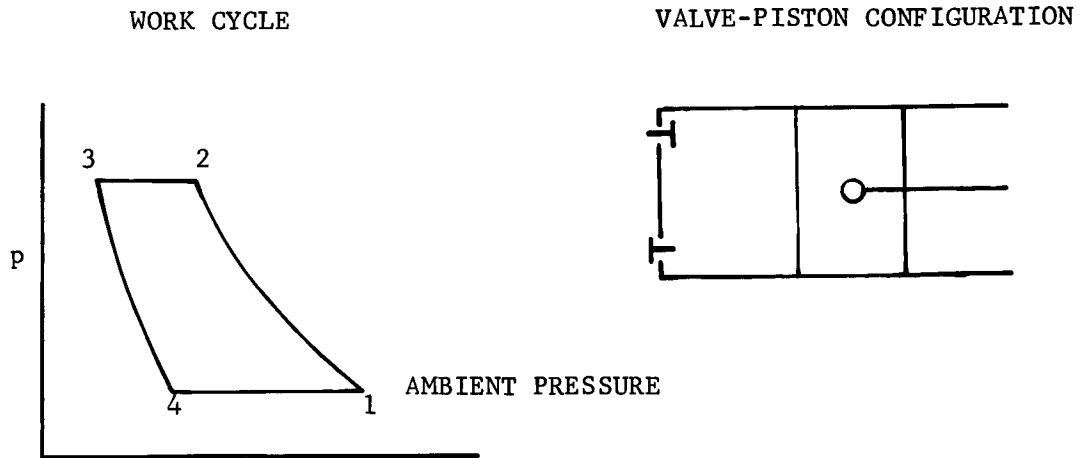


FIGURE 2-36. OPERATIONAL CYCLE AND VALVE-PISTON CONFIGURATION FOR A CONVENTIONAL PISTON COMPRESSOR

In this cycle, the following steps occur.

- (1) Gas is admitted at constant pressure from point 4 to 1 as the piston approaches bottom dead center.
- (2) The air is compressed isentropically from point 1 to 2 at which point the spring-loaded valve opens.
- (3) Compressed gas is discharged at constant pressure as the piston approaches top dead center.
- (4) The residual gases are expanded until ambient pressure is reached at which point the intake valve opens and the process repeats itself.

In order to compare the previous compressor with this more conventional one, the power expended was calculated. The same piston size, exhaust pressures, and compressor speeds were assumed. The work required per cycle was used as the means of comparison. It was found that the compressor with conventional valving required less power per cycle in the ratio of 2.5 to one for the same weight of delivered compressed gas. Thus, the conventional valving for a piston compressor is more efficient from an energy consumed viewpoint.

It was desired to estimate the power requirements for a conventional piston compressor that could deliver 50 cubic feet of free air flow under the conditions of the 10 millibar atmosphere. From Reference 6, the work for a conventional piston compressor in terms of compression ratio is given by

$$\text{Work} = \frac{k\dot{W}RT_1}{1-k} \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right]. \quad (20)$$

From this relation, it is seen that the work expended in compressing a gas is reduced as the compression ratio is decreased since the term in the brackets approaches zero. The variation in power as a function of compression ratio is shown in Figure 2-37. In Section 3.3 it is shown that a pressure ratio up to 1.3 to 1 is required to produce flow velocities of 40 feet per second in a half inch diameter tube 10 feet long. For these low compression ratios the power required to compress the gas is almost negligible in the order of 1 to 2 watts. The power lost in mechanical friction will probably exceed this power by an order of magnitude. Thus, a positive displacement compressor will work in a Martian atmosphere with small power input requirements; however, the size of the compressor will be relatively large and bulky.

2.3.3 SUPERSONIC JETS

In generating the design concepts contained in Appendix B, several of these incorporated high speed or supersonic jets. The supersonic nozzle is mechanically very simple and the appealing aspect of a high speed jet of air for soil sampling is the ease with which it can penetrate crevices and accommodate itself to uneven terrain conditions. The high velocity of a supersonic flow should be very effective in lifting loose surface material into the air. If this dust cloud is contained inside a hood or shroud, it should be a very effective means of obtaining a sample. Also a contained dust cloud surrounding a jet should cause some of the soil particles to be accelerated to impact the surface thereby eroding some of the cohesive surface material from the soil.

To estimate the flow characteristics for a small supersonic nozzle, the nozzle geometry described in Figure 2-38 was assumed. A converging entrance radius equal to three throat radii is followed by a simple diverging cone with a cone half angle of 10 degrees. The pressure variation along the nozzle for an assumed chamber pressure of 10 psi is shown in Figure 2-39. It is seen that expansion is fairly well completed for a nozzle length which exceeds 10 throat radii. Since higher chamber pressures may be desired, the variation of expansion ratio required to expand to a 10 millibar ambient pressure at the exit was estimated and is shown in Figure 2-40. Expansion ratios up to 30 may be required. The calculated velocity variation is shown in Figure 2-41.

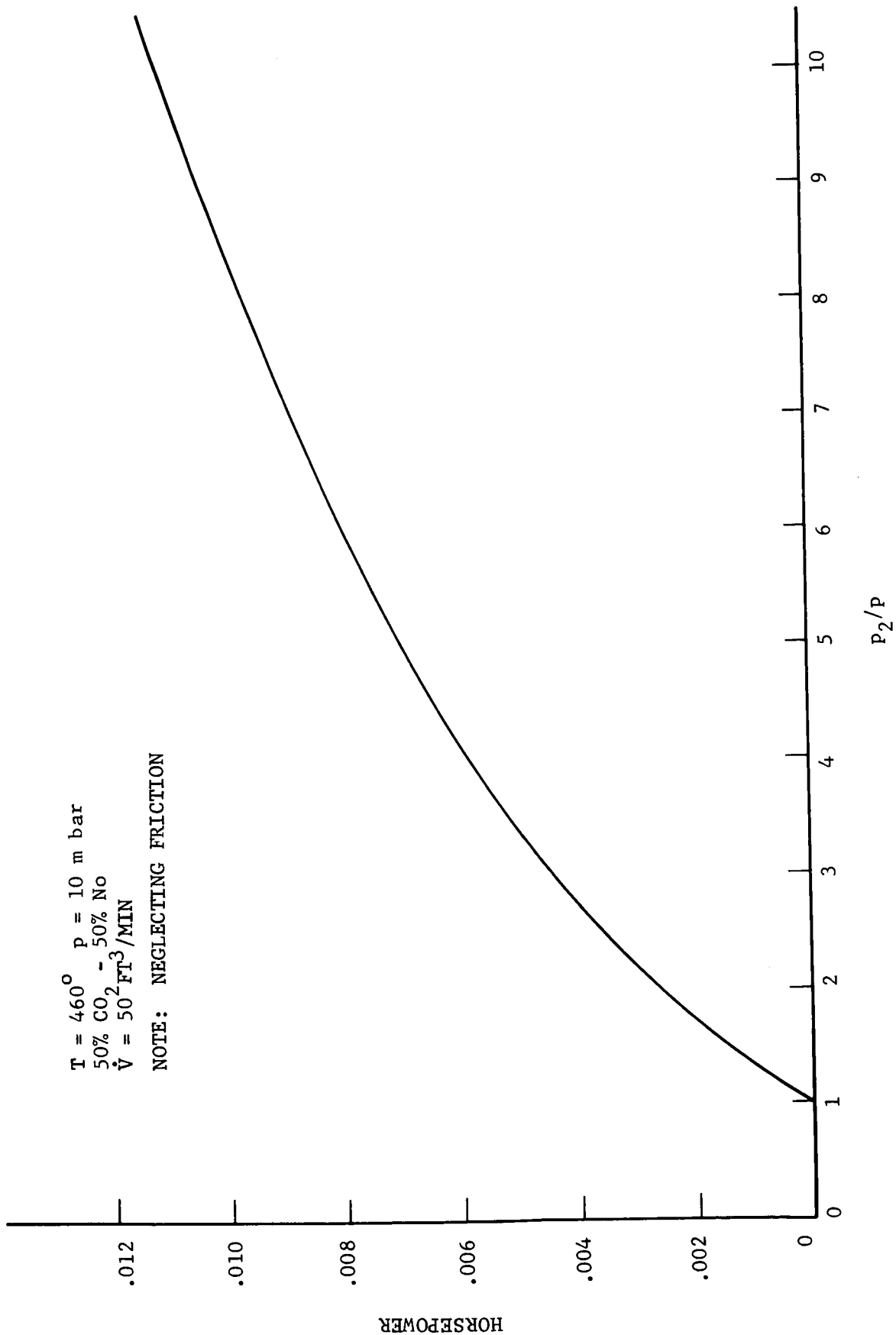


FIGURE 2-37 HORSEPOWER VARIATION WITH COMPRESSION RATIO

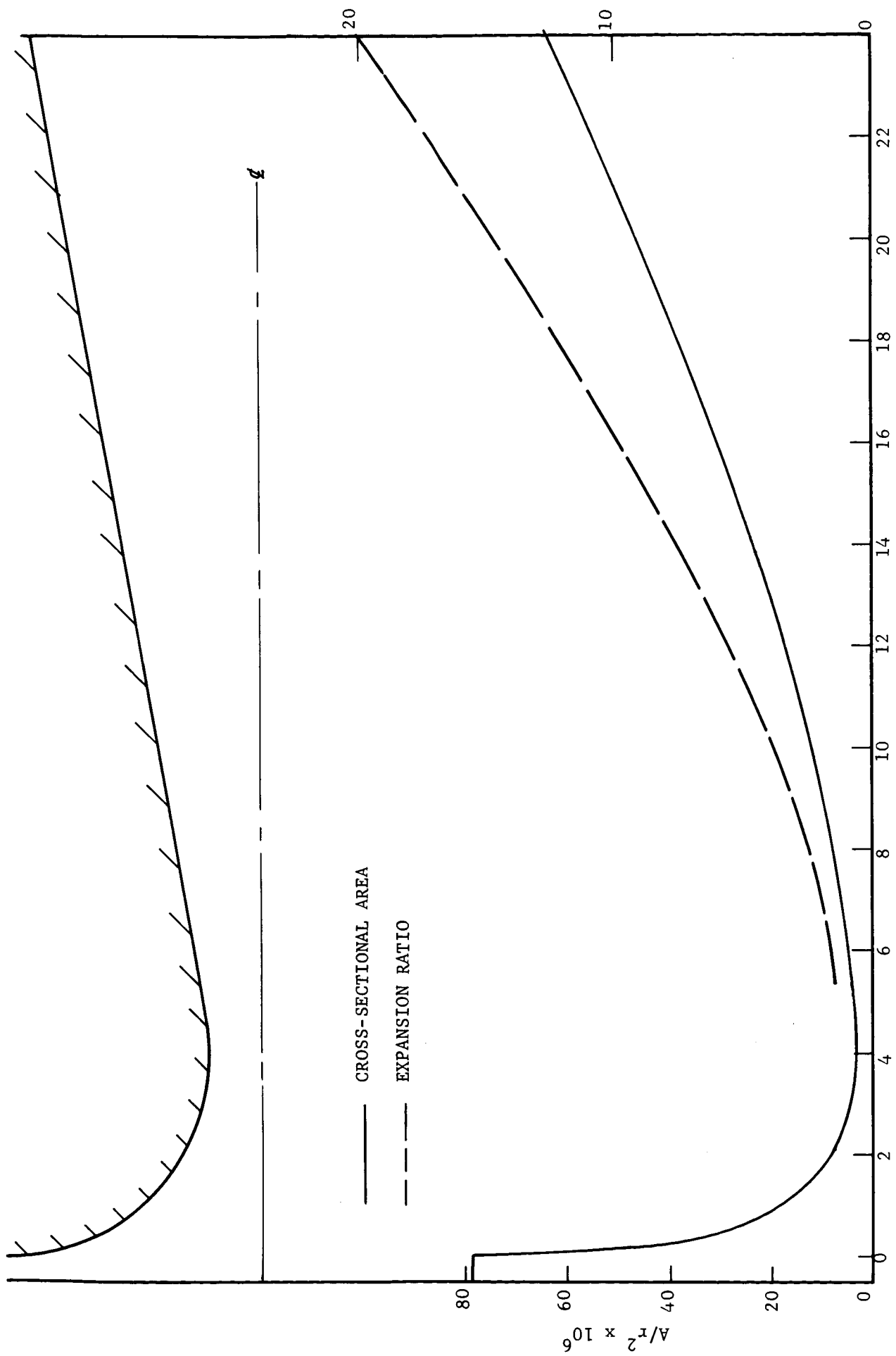


FIGURE 2-38 ASSUMED NOZZLE CONFIGURATION

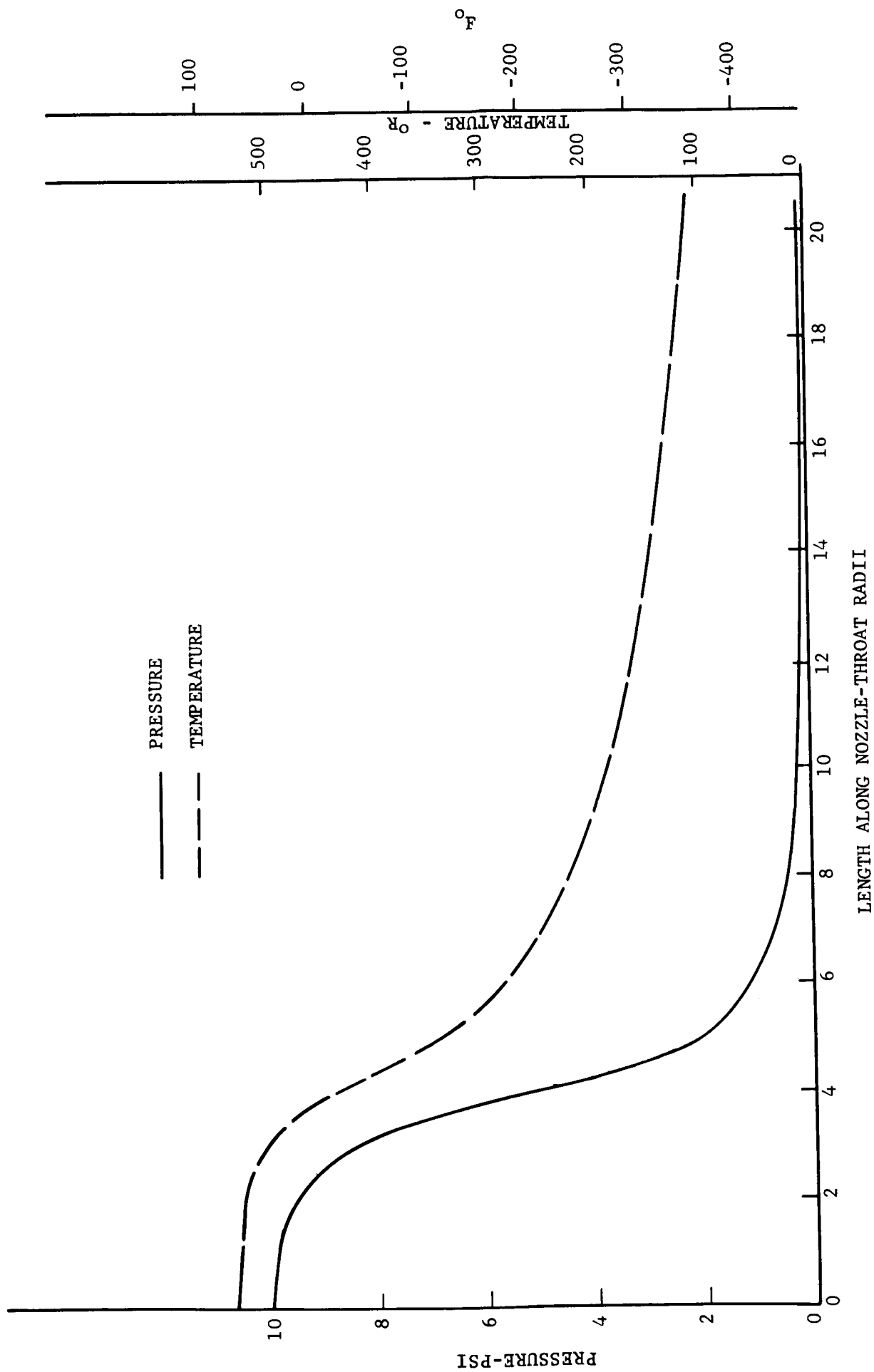


FIGURE 2-39 PRESSURE VARIATION ALONG NOZZLE AXIS

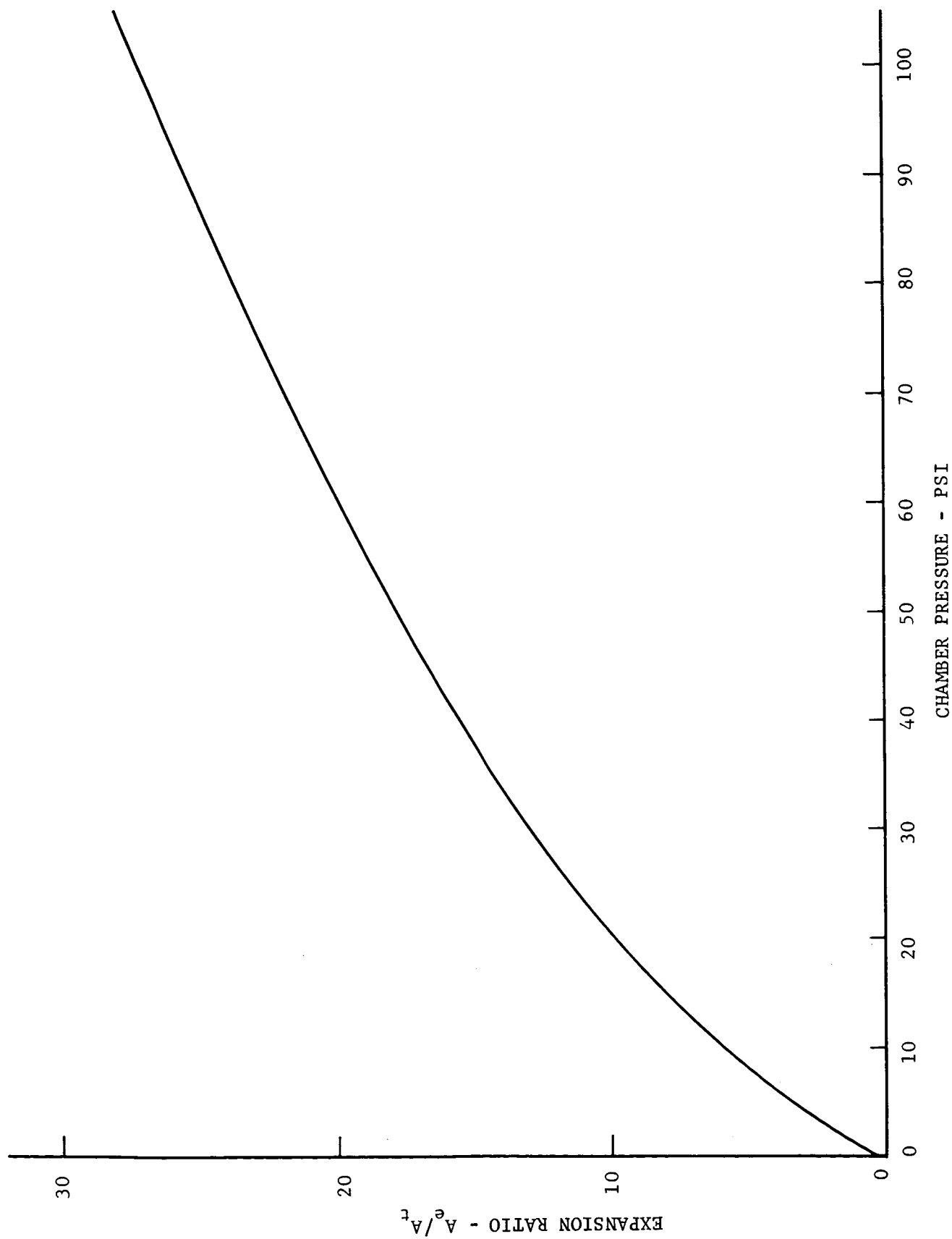


FIGURE 2-40 EXPANSION RATIO REQUIRED TO EXPAND TO 10 MBAR AMBIENT PRESSURE

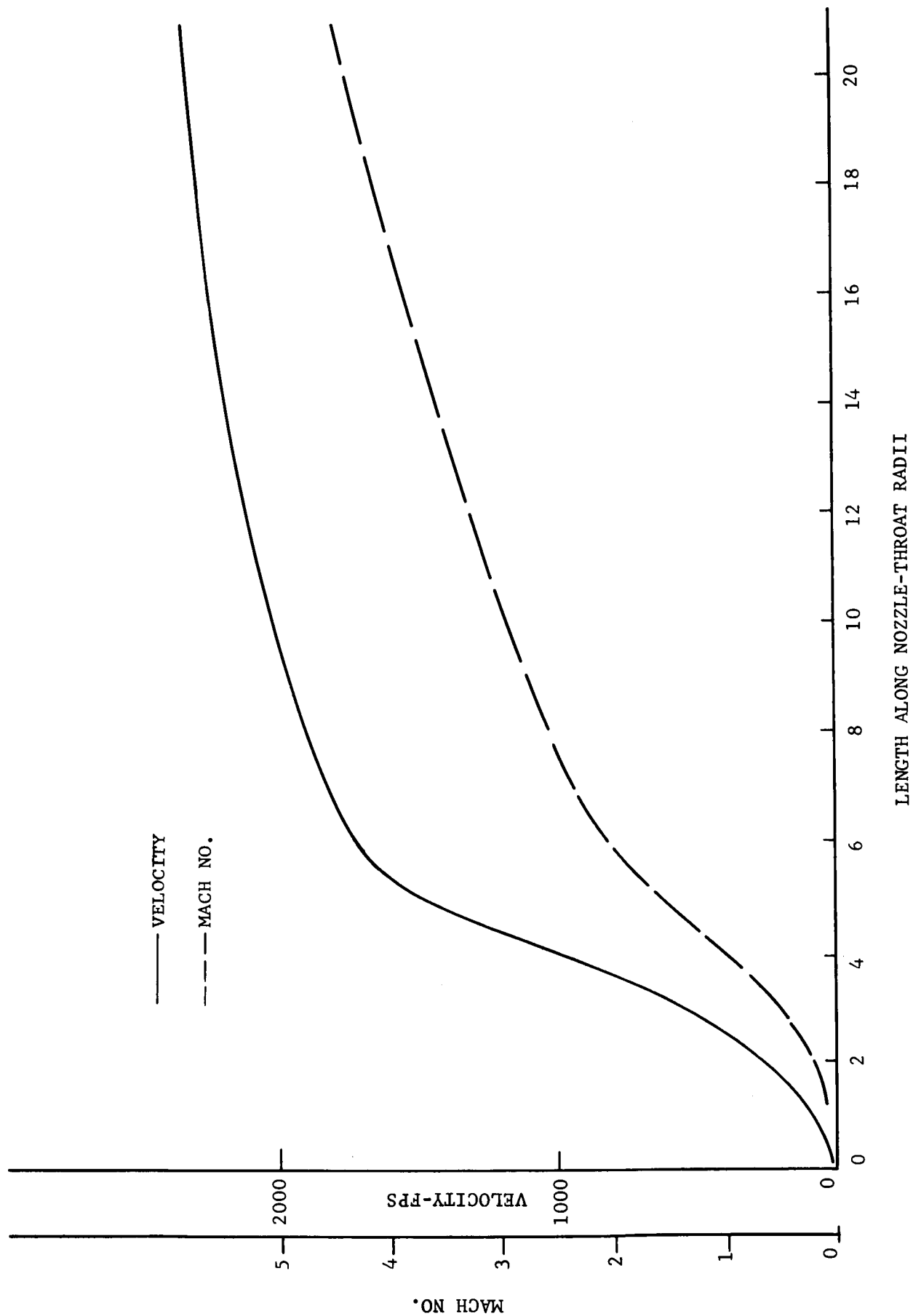


FIGURE 2-41 VELOCITY VARIATION ALONG NOZZLE AXIS

It is seen that exit velocities in the order of 2000 feet per second can be expected.

The mass flow rate for this nozzle configuration was determined as a function of throat diameter for two chamber pressures of 10 psi and 100 psi, which is shown in Figure 2-42. It is seen that flow rates do not become excessive for the lower pressures. The optimum chamber pressure will probably have to be determined experimentally. The higher chamber pressure requires a larger expansion ratio which in turn will produce a larger diameter jet. The larger jet should be less susceptible to attenuation of the velocity and will cover a larger area which is desirable.

This effort was very limited in scope and was terminated at this point since none of the concepts involving aerosolizing jets were selected for further development, however, an interesting phenomenon involving multiple nozzles was reported in NASA TN D-1017 by A. A. Spady, Jr.¹⁰ This work was an empirical study to determine the characteristics of the dust cloud raised by a vehicle landing on the lunar surface. Various configurations of nozzle clusters were tested in a vacuum over a surface covered with fine particles to simulate dust. In the four nozzle configuration Spady discovered that a column of dust formed in the center of the four nozzle pattern and extended up to the nozzle mounting block, as shown in Figure 2-43 taken from the technical report. This suggests that a multiple nozzle arrangement could probably be optimized to preferentially concentrate the dust cloud in an area such as a pneumatic transport tube leading to the soil sample collector, thereby enhancing the soil collection rate.

2.3.4 OPERATIONAL STUDIES AND CONCLUSIONS

The conceptual designs for sample handling systems were to be developed with certain specific requirements and design goals in mind. In addition to these requirements and goals, a series of factors were to be considered in the design. These factors included:

- (1) Survival of impact from a hard landing capsule at, for example, 2000 to 10,000 g.
- (2) Sterilization requirements in accordance with Voyager specifications.
- (3) An operating temperature range taken as -100°F to 150°F.
- (4) The possibility of a need to penetrate a considerable thickness of impact absorbing material.
- (5) The possibility of acquiring a sample from a non-gravity oriented capsule.

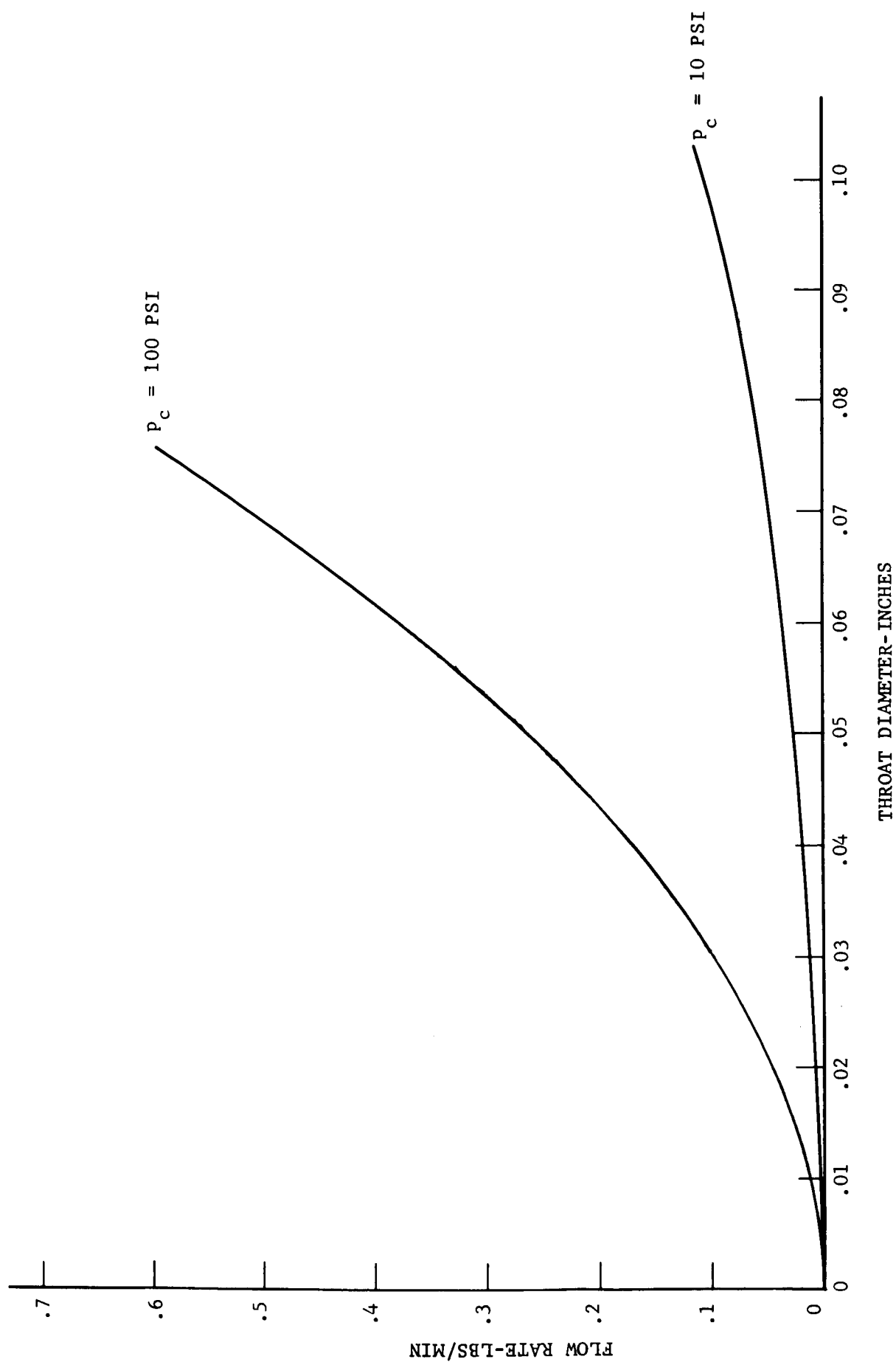


FIGURE 2-42 MASS FLOW RATE THROUGH A SUPERSONIC NOZZLE

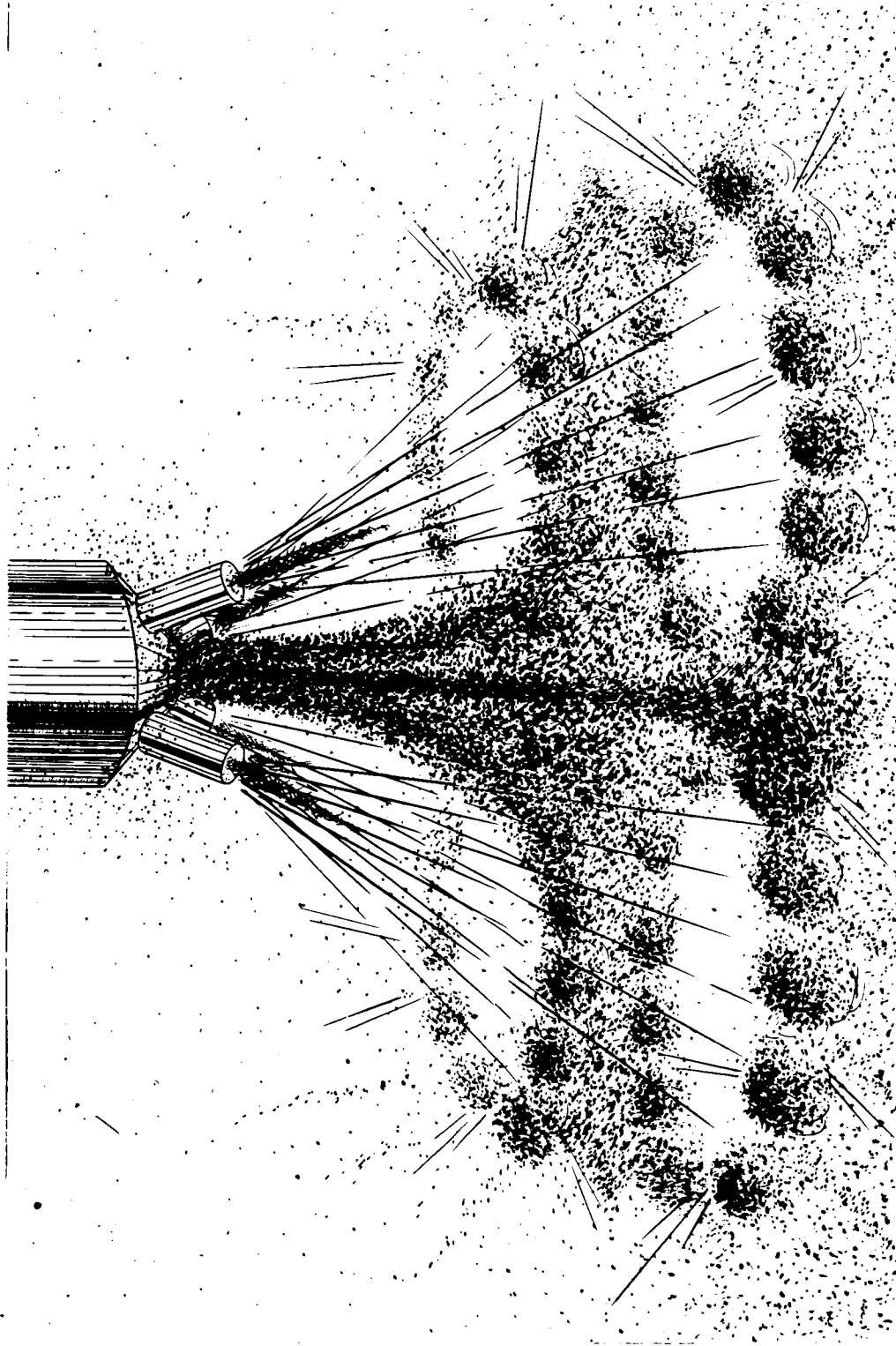


FIGURE 2-43 EXAMPLE OF FOUR NOZZLE FLOW PATTERN ON DUST COVER SURFACE

These considerations were not established as rigid constraints, merely design guidelines.

Impact hardening of mechanical systems is basically a packaging problem provided that the systems are slightly overdesigned from a strength of materials standpoint. The mechanical systems contained within the prototype samplers lend themselves directly to impact hardening for flight hardware, especially if the impact event is unidirectional. If the impact is omnidirectional, the packaging and hardening problem will be more severe. In the interest of developing prototype hardware rapidly, the hardening consideration was not emphasized.

Sterilization requirements can be met by utilizing sterilizable motors or pneumatic systems, silicon and molybdenum sulfide lubricants, and inorganic temperature-tolerant bushings, washers, bearings and seals. Except for motors, the choice of materials, in most cases, was made with the sterilization requirements in mind.

The operational temperature range set at -100° to 150°F should pose no problem when sterilization requirements are met and the operating environment is assumed to be exceptionally dry.

The possibility of a need to penetrate a considerable thickness of impact limiting material has received a large amount of study at Aeronutronic. In the first place, it is recognized that bio-capsule impact limiters will have to be designed from metal-honeycomb to avoid organic contamination from balsa or phenolic-honeycomb limiters. The removal of a metal-honeycomb limiter from the instrument payload proper or from critical areas of the payload must be accomplished without injuring or contaminating the payload or loading the nearby soil with extraneous chemicals. First, it is necessary to separate the crushed limiter into segments so they can be ejected from the payload. Ordinary linear or shaped pyrotechnic charges do not appear feasible for this purpose from the contamination standpoint. Therefore, we are left with the following choices for separating the limiter into segments:

- (1) Cutting or sawing operations.
- (2) Disconnection of existing segments by solenoids, pneumatic cylinders, "Pyrofuse," or sealed pyrotechnic pin pullers.

To separate the limiter segments from the payload, we have the following possibilities to choose from:

- (1) Springs
- (2) Pneumatic cylinders

(3) Pyrotechnic guns

(4) Gravity

The problem of non-gravity orientation has also been studied. These studies indicated that in capsules up to three feet in diameter, liquid floatation is a practical way to achieve gravity orientation. This system utilizes an internal payload which is floated with neutral buoyancy within an outer shell. The density of the flotation fluid must be equal to the density of the payload. The center of gravity of the payload sphere must be below the geometrical center of the sphere. After landing, the gravitational force is allowed to erect the payload to a vertical position; the payload is then locked to the outer shell. This method implies the use of stabilizing legs attached to the outer shell. Two alternate concepts can be advanced for larger capsules. In the first, the capsule is shaped for unidirectional stability. If a spherical payload must be delivered, then large orange peel-shaped segments of the impact limiter can be released and rotated about a point on the south pole of the sphere to effect a simultaneous erection and stabilization of the payload. In the event that the vehicle were to come to rest directly on the north pole, a foot or instrument mast could be extended simultaneously to roll the vehicle on its side and insure gravity stabilization from any random orientation. In other words, it appears to be more logical to orient the capsule in the first place rather than attempt to design instruments that will effectively operate in any random direction.

SECTION 3

BREADBOARD TEST FIXTURE PHASE

Prior to the conclusion of the Conceptual Design Phase of the program certain subsystems that were common to the semifinal list of sample handling systems appeared to merit testing by means of simple breadboard test fixtures. These included pneumatic tubes and telescoping booms. By this point in time, the boom deployed wire brush sampling head was being recommended by Aeronutronic personnel despite several operational questions. In order to supply some rapid but effective answers, the sampling head and boom test fixtures were fabricated and tested on soil models in rotating bins. The experience gained by operating and testing these breadboard fixtures was invaluable for final prototype design and fabrication purposes.

During this phase the soil models stipulated in the contract were collected and/or manufactured. Rotating bins for testing soils with the boom deployed wire brush test fixture were designed and fabricated. Gallon-size tin cans were obtained and filled for future vertically deployed sampler testing.

3.1 SOIL MODELS AND TEST FACILITIES

3.1.1 DEFINITION OF SOIL MODELS

Dr. Douglas B. Nash, geologist, and Dr. Gerald A. Soffen, microbiologist, of JPL at the request of the JPL Soil Sampling Laboratory, defined six soil models that should span the general post-Mariner IV range of the uppermost surface materials expected to be encountered on Mars. These models are identified in Table 2-1. The following four models were selected from this list and incorporated within the Supplemental Agreement:

- (1) Cohesive powder
- (2) Cohesionless particulate
- (3) Hardpan
- (4) Loose rubble

The stipulated models were related to a grading analysis performed previously on the ABL study.¹¹ In this study soil particle size distributions from the soil mechanics and natural soil literature had been compared and were found to range close to either of two representative values of standard deviation for an idealized normal or Gaussian distribution. The representative values determined were approximately 0.28 and 0.72. The high standard deviations are characteristic of agricultural soils which consist of agglomerations of finer material bound together to form large particles. The agglomerated particles do not usually break down into the finer particles unless they are dispersed, for example in an NaOH solution. From the data evaluated, the sands and gravels tend to fit the lower standard deviations which can nearly always be linked to some form of natural grading action such as wind or water transport. On the basis of these results, the lower standard deviation appears to be the better choice for a postulated Martian soil and is therefore used as a basis for defining the soil test models. On this basis, two representative grain size distribution curves have been selected to identify the finer particulate models, sand and silt. These are shown in Figure 3-1. The solid circles are points representing mean and extreme diameters. The slope of the solid lines corresponds to a standard deviation of 0.28 as determined in the previous ABL study. These distributions represent a reasonable idealized fit to a normal distribution for purposes of defining soil test models. It is noted that the end points of the size range are not sharp cutoff values, since the normal distribution curve approaches zero asymptotically. Thus, variations from the assumed curve at the extremes of the range of sizes are less significant. The unsorted rubble model does not conform to this treatment; see the description below for its composition and character.

The following outline describes the specific materials selected, their composition, grain size range and physical character:

Model No. 1 Cohesive powder

Composition: Compacted, crushed, very angular grained, olivine basalt silt having the following composition:

Plagioclase (feldspar)	40%
Pyroxene	40%
Olivine	15%
Magnetite	5%

Grain Size: The approximate grain size distribution is 100% through a No. 325 standard sieve ($<44\mu$).

Grain size range:	100% $<44\mu$, 90% $>15\mu$ and 10% $<1.5\mu$
Mean grain size:	32μ (medium-grained silt)

Physical character: For testing purposes this model is prepared in a cohesive state. A high degree of cohesion is obtained through a combination of extremely angular grain shape and vibratory packing.

Model No. 2 Cohesionless particulate (sand)

Composition: Nevada No. 60 mechanically graded, rounded to subrounded grained, natural desert dune, quartz glass sand with the following grain size distribution:

Grain size:

0.6% retained on a No. 30 standard sieve
7.7% on No. 40 (420μ)
18.3% on No. 50 (297μ)
29.0% on No. 70 (210μ)
33.7% on No. 100 (149μ)
9.0% on No. 140 (105μ)
1.77% through No. 140 ($<100\mu$)
Grain size range: 91.6% $<300\mu$ and 89.3% $>150\mu$
Mean grain size: 250μ (medium grained sand)

Physical Character: For testing purposes this model is left in a loose state. Since the grains are well-rounded to subrounded an easily sheared uncompacted product is obtained.

Model No. 3 Hardpan

Discussion: In order to produce the effect of cementation, four submodels were constructed to yield models having an order of magnitude increase in shear strength response to the degree of cementation and, respectively, simulate:

- (1) Soluble salts duricrust (25 psi shear strength)
- (2) Friable hardpan (250 psi shear strength)
- (3) Hardpan (2500 psi shear strength)
- (4) Natural adobe hardpan

Composition: Ottawa No. 398 flour, a mechanically graded, subrounded grained, natural fossil beach source, quartz glass flour or coarse silt having a mean grain size of 40μ . The Ottawa silt was bonded with materials identified below. The natural Mexican adobe brick is manufactured from native adobe mud whose grain size varies from clay ($<1\mu$) to coarse sand (2mm) and a few small pebbles ($>2\text{mm}$).

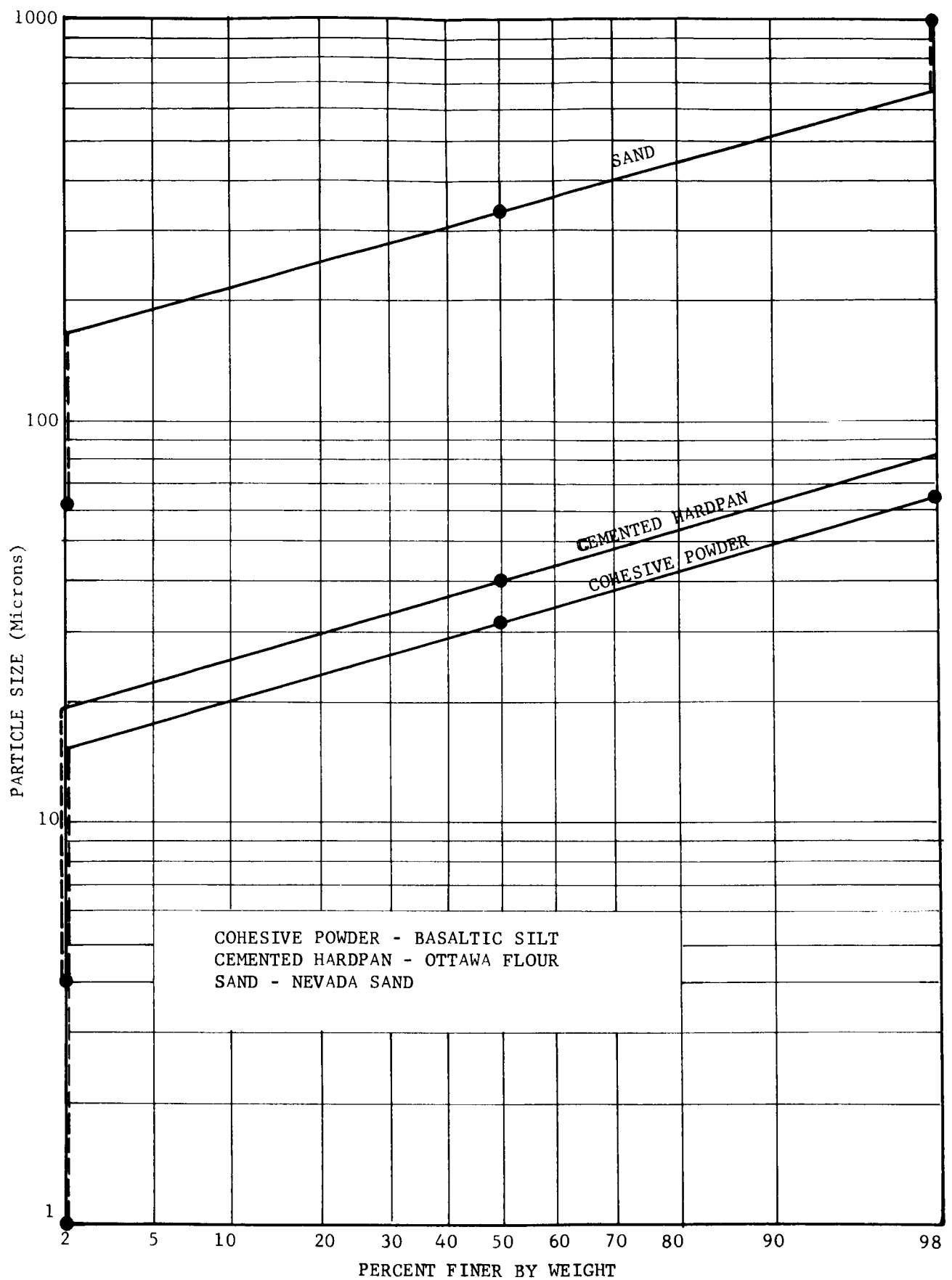


FIGURE 3-1 GRAIN SIZE DISTRIBUTIONS FOR TEST SOIL MODELS

Preparation:

Submodel No. 3a: Bond silt by introducing a sufficient quantity of warm supersaturated salt water to completely cover silt-filled bin. Evaporate model to dryness.

Submodel No. 3b: Bond silt with a 7.5:1 ratio of silt to common cement, fill bin and cure.

Submodel No. 3c: Disperse a 10:1 ratio of sand and polyester resin (Glass Fiber Products Co. PE-9538) in bin and cure.

Submodel No. 3d: Natural adobe brick; Adobler's kiln-dried (wood fired) adobe bricks.

Testing: Submodels 3a, 3b and 3c were cast into standard cylinders, 2 inches in diameter and $3\frac{1}{2}$ inches long, for compression testing. The failure stresses and failure modes are identified in Table 3.1.

TABLE 3-I

SIMULATED CEMENTED HARDPAN TEST DATA

<u>Submodel No.</u>	<u>Failure Stress, psi</u>	<u>Cleavage Angle, θ</u>
3a	23	35°
3b	200	20°
3c	3400	40°

An approximate order of magnitude spread in strength values was achieved as intended. Submodels 3b and 3c are not affected by moisture whereas Submodel 3a is affected. To maintain control of the properties for 3a, this formulation should probably be baked immediately prior to use to dry it out. Based on the above described tests and test discussion, it was decided that Submodel 3b would be the ideal test model for cemented hardpan. For sampler testing purposes, this model was specifically prepared as follows:

<u>Material</u>	<u>Description</u>	<u>Weight</u>	<u>Remarks</u>
Sand	Nevada No. 60	34.85 lb.	Ratio of aggregate/cement = 7.36:1 by weight
Silt	Ottawa No. 398 Flour	33.08 lb.	
Dye	Powdered Fe_2O_3 ($<44\mu$)	0.88 lb.	
Cement	Common Portland	9.29 lb.	
Water		15.00 lb	

The dry material was completely admixed, water added, and mixture stirred until the consistency of plaster-of-paris was attained. The slurry was poured into the standard test bin to $3/4$ of its total capacity and restirred to eliminate most of the air bubbles. It was cured for approximately 2-3 weeks prior to testing.

Model No. 4 Rubble

Discussion: Two rubble models were prepared; one composed of randomly mixed rubble, and another to model a desert pavement type of surface.

Composition: A mixture of equal parts by weight of the following materials:

- (1) Hand picked, naturally comminuted cobbles composed of vesicular olivine basalt having a grain size range of approximately 50-125 mm.
- (2) Hand picked, naturally comminuted and/or crushed vesicular olivine basalt pebbles with a grain size range of 2-25 mm.
- (3) Crushed olivine basalt and sand having a grain size range of 64μ to 2mm .
- (4) Crushed basalt silt with a grain size range of 1.5 - 44μ .

Preparation:

Submodel No. 4a (randomly mixed model): Fill bin with randomly mixed rubble.

Submodel No. 4b (desert pavement model): Fill bin $3/4$ full with loose basalt silt, sand and pebbles mixture; hand emplace a mosaic of basalt cobbles and cover with thin layer of loose mixture; force mosaic into loose mixture as far as possible; finish by blowing air jet across surface of model.

3.1.2 DESCRIPTION OF TEST FACILITIES AND ACCESSORIES

Test facilities and accessory devices had to be prepared in order to test breadboard models of the rotating wire brush and the pneumatic transport tubes. Rotating test bins were prepared to test the wire brush sampler head. A manometer was constructed in order to determine the pressure drop and flow velocity in a pneumatic tube. The NASA Mars simulation chamber was utilized to test the soil transport capabilities of pneumatic tubes under reduced air pressure conditions.

The rotating soil test bins, pictured in Figure 3-5, are 6-sided wooden bins having the dimensions, 2' diameter x 6" deep. The base is a round plywood plate revolving on 4 rubber tired casters. The plate is friction driven by an electric motor to revolve at speeds up to 2 rps. In order to compact soils, four wedge-shaped metal strips were attached to the base plate. Each strip is $1/8$ " thick so that as the bin is revolved on the four supporting casters, 16 drops of $1/8$ " are acquired per revolution. At 2 rps, the soil is compacted at 32 cps, approximating the optimum compacting rate of 28 cps. The compacted silt model (Model No. 1) was prepared in this fashion. The cohesionless particulate material (Model No. 2) was merely poured into the bin and the surface leveled prior to testing. The preparation of the cemented hardpan model (Model No. 3b) and the desert pavement model (Model No. 4b) was described above. The model is illustrated in Figure 3-5. The natural adobe brick model, Model 3d, consists simply of three 12×1 " adobe bricks sawed to conform to the geometry of the floor of a test bin and emplaced thereon.

At this time, in order to insure soil model uniformity, one-gallon tin cans were filled two-thirds with cemented hardpan for future testing of the vertically deployed conical sieve sampler.

An inclined manometer, pictured in operation in Figure 3-30, was constructed to measure the velocity profile, pressure drop and flow velocity existing within the pneumatic tube breadboard fixtures tested. The inclined manometer was fabricated with a slope calculated to give an order of magnitude magnification of pressure readings. It can be read to a 0.1 inch which provides a measured accuracy to 0.01 inches of water.

The NASA Mars simulation chamber, I.D. No. NASA-100-3, was received from Litton Systems, Inc. on 12 July 1966. The simulator is 2 feet in diameter and 5 feet long. The exterior is enclosed in a rectangular plywood box containing thermal insulation. The access door consists of a hinged end dome, also insulated. There are eight viewing windows in the chamber. Two are located on each side, two in the fixed dome, one on top, and one in the door of the chamber. Seven 5/16-inch diameter tubing feed throughs and three 1/4-inch pipe feed throughs are provided. The electrical feed through consists of a 19-pin connector. Two 110 volt receptacles are provided inside the chamber. The chamber contains no means for circulating the internal atmosphere. It is our understanding that Litton Industries used a 90 cfm vacuum pump with the chamber. To chill the chamber to -110°F , Litton used an alcohol/dry ice slurry which was pumped through the cooling coils wrapped around the cylindrical part of the chamber. The simulation chamber was delivered without the vacuum pump and the heat exchanger pump for the alcohol slurry.

The vacuum pump used at Aeronutronic has a 15 cfm capacity which was used to evaluate the performance of the chamber. The chamber pump down rate achieved with this pump is shown in Figure 3-2. The pump down rate is seen to be linear at 4.6 mm/minute down to the pressure of interest in this program. A minimum pressure of 300 microns can be achieved. The leak rate was checked by pumping to the minimum pressure and recording the pressure rise as a function of time. In the region of interest (5 to 10 mb) the leak rate is very nearly linear and has an average value of 0.193 mm/minute, as shown in Figure 3-3. Under room temperature conditions the pumping rate exceeds the leak rate by a factor of 24. Since no pump system for chill down was readily available, liquid nitrogen was vented through the cooling tubes bonded to the chamber. It was established that this method can be used to achieve a temperature of -100°F in approximately 1.5 hours using about 44 litres of liquid nitrogen.

The chill down characteristics with an ambient pressure of 1 atmosphere in the chamber is shown in Figure 3-4. To evaluate the combined performance of chill down and low pressure, the chamber was pumped down to 300 microns operated continuously during chill down. As the temperature decreased, the chamber pressure increased at a continuously higher rate until the vacuum

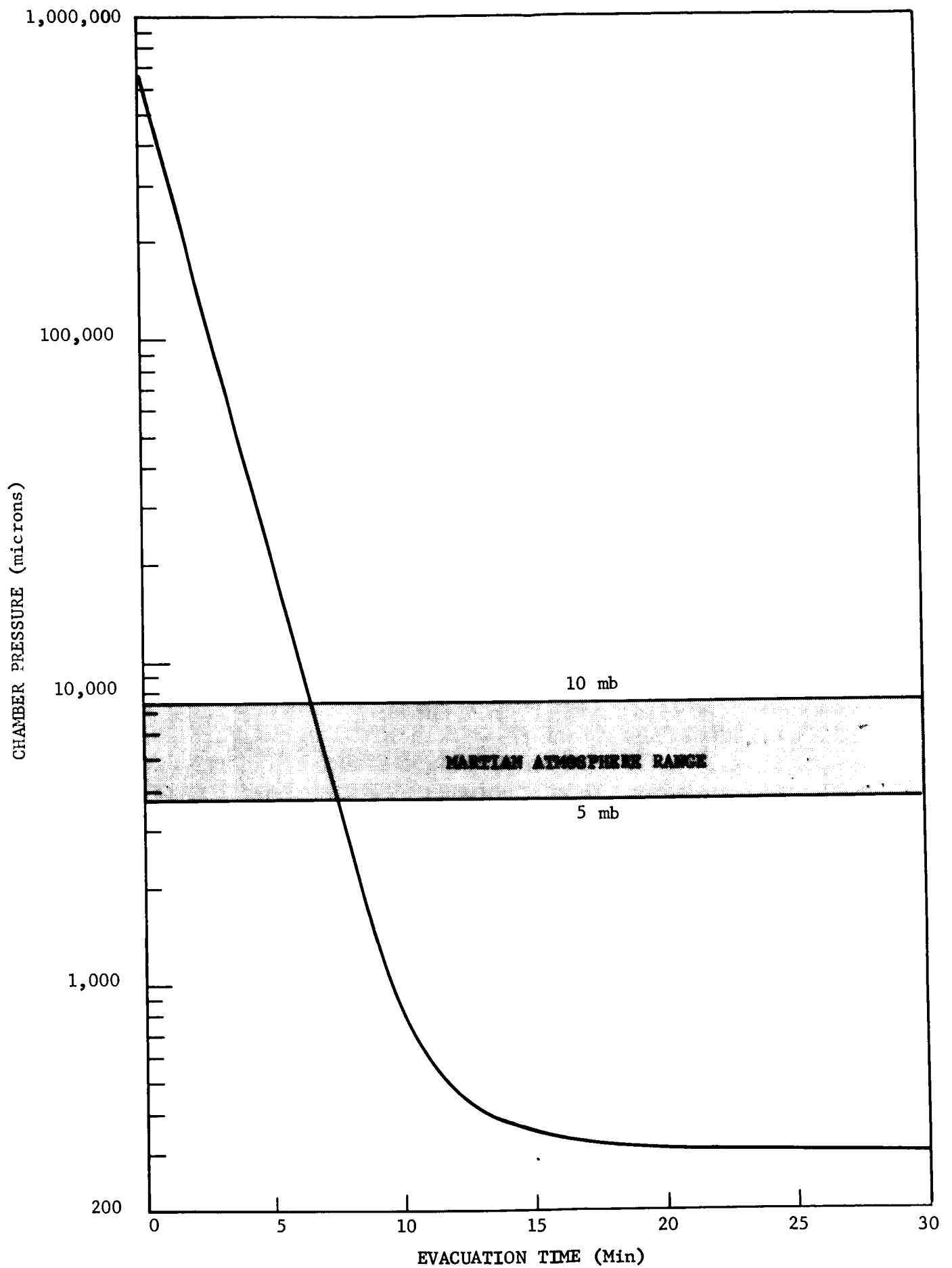


FIGURE 3-2 SIMULATION CHAMBER PUMP DOWN RATE

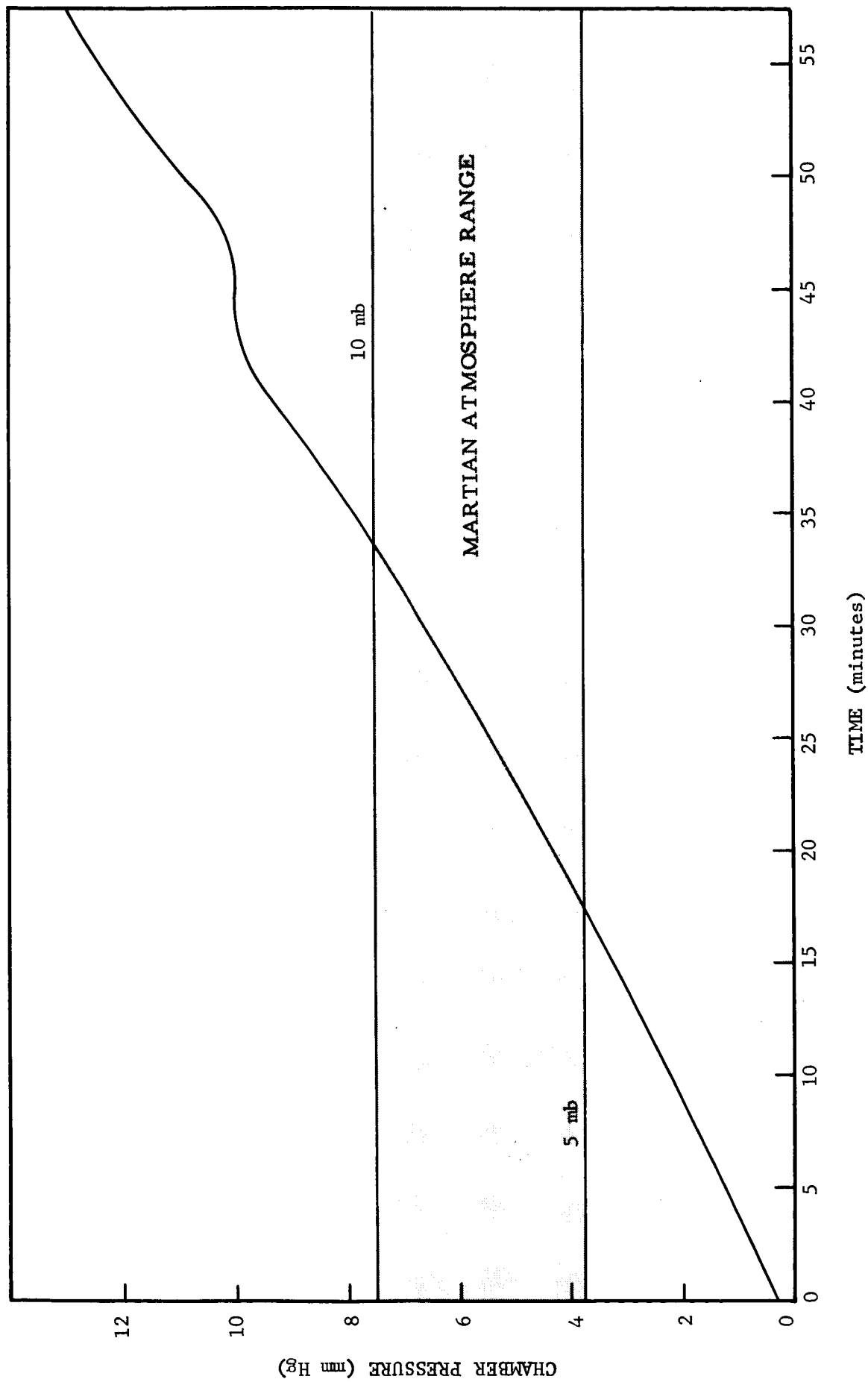


FIGURE 3-3 SIMULATION CHAMBER LEAK RATE

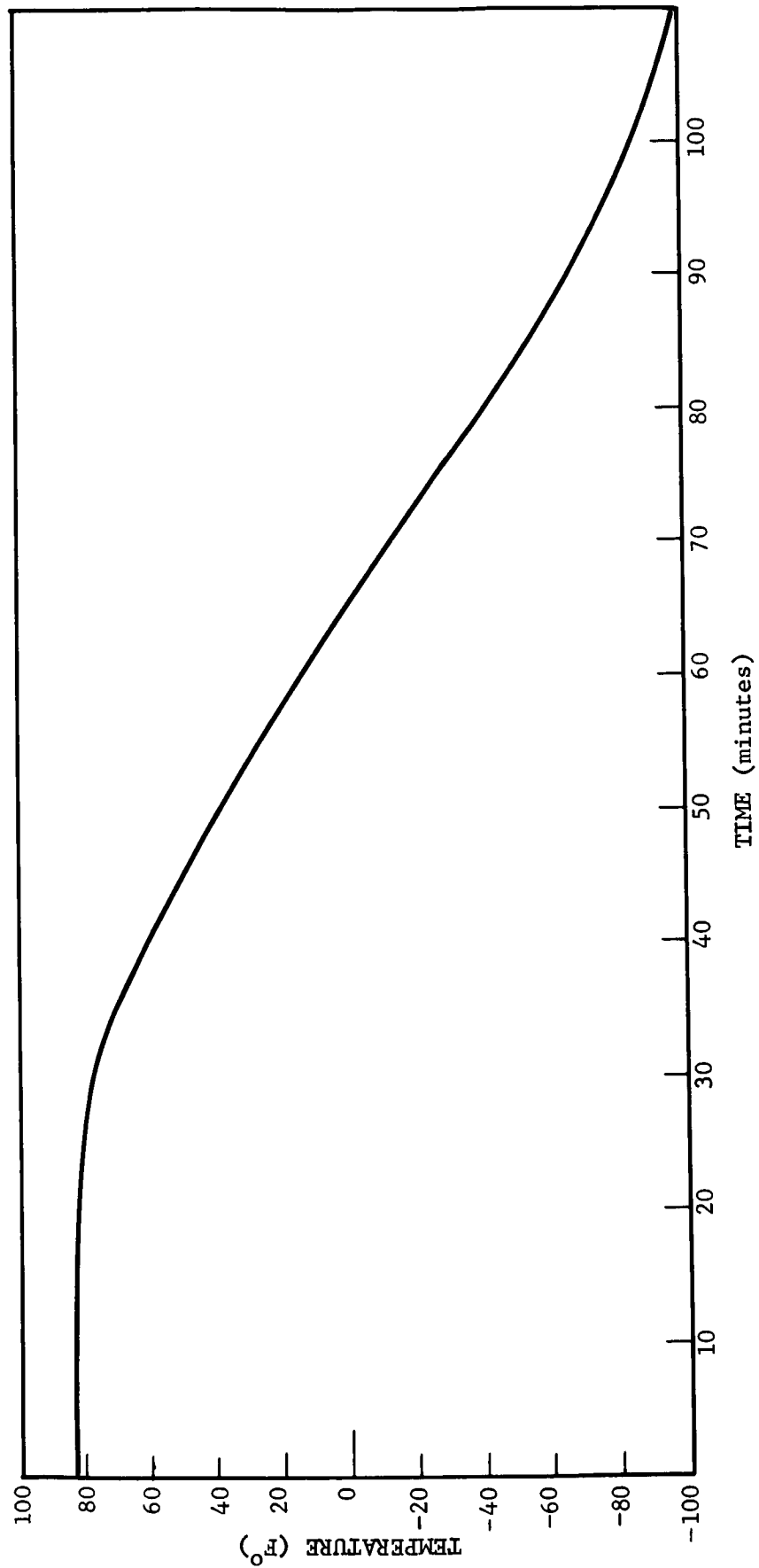


FIGURE 3-4 SIMULATION CHAMBER CHILL DOWN CHARACTERISTICS

seal was completely lost and the chamber returned to atmospheric pressure. The loss of seal occurred at the chamber access door seal. A detailed examination revealed the following defects in chamber door construction.

(1) The door is sealed by one "O" ring which appears to be undersized in cross-section to be compatible with the "O" ring groove in the door.

(2) The "O" ring groove contains machine marks of sufficient depth to make an efficient seal doubtful.

(3) The chamber surface against which the "O" ring seats contains two holes which have been plugged and present an irregular surface in the area of "O" ring contact.

(4) The plywood door exterior bears against the plywood exterior of the chamber possibly limiting the compression of the "O" ring seal.

(5) The door hinges are rigid which may also limit "O" ring compression in the area of the hing.

(6) The use of the rigid door hinges also causes a wiping action on the "O" rings seal as the door is closed which tends to remove sealing compounds such as high vacuum silicon grease.

The chamber sealed properly after this check when the chamber temperature had returned to ambient room temperature. Thus, the above mentioned factors coupled with thermal distortion during chill down probably accounts for the loss of the vacuum seal when combined low pressure and low temperature operation is attempted.

3.2 ROTATING WIRE BRUSH BREADBOARD

It was felt that a sufficiently large number of variables and unknowns existed in terms of the interaction between the wire brush, the brush hood or shroud, and the soil model being sampled to warrant some breadboard development efforts to help determine the correct configuration. The primary objective of this breadboard testing was to determine the relative merit of a floating hood as opposed to a fixed hood, to develop the internal configuration requirements of the hood, and to determine the relative efficiency of mechanical sample collection as opposed to pneumatic collection.

Since the wire brush sampler was to be boom mounted and would be sampling while traversing the surface, rotating soil sample bins were fabricated to provide the correct relative velocity of one foot per second between the wire brush and the soil surface. A typical sample bin is shown in figure 3-5 with the rotating wire brush breadboard in the floating hood configuration. The wire brush breadboard was supported on the end of a fixed boom pivoted at the center and counterbalanced at the end. The total breadboard setup is shown in the photograph in figure 3-6. This arrangement allows the angle of attack of the sampling head and the normal force acting on the wire brush to be adjusted as desired.

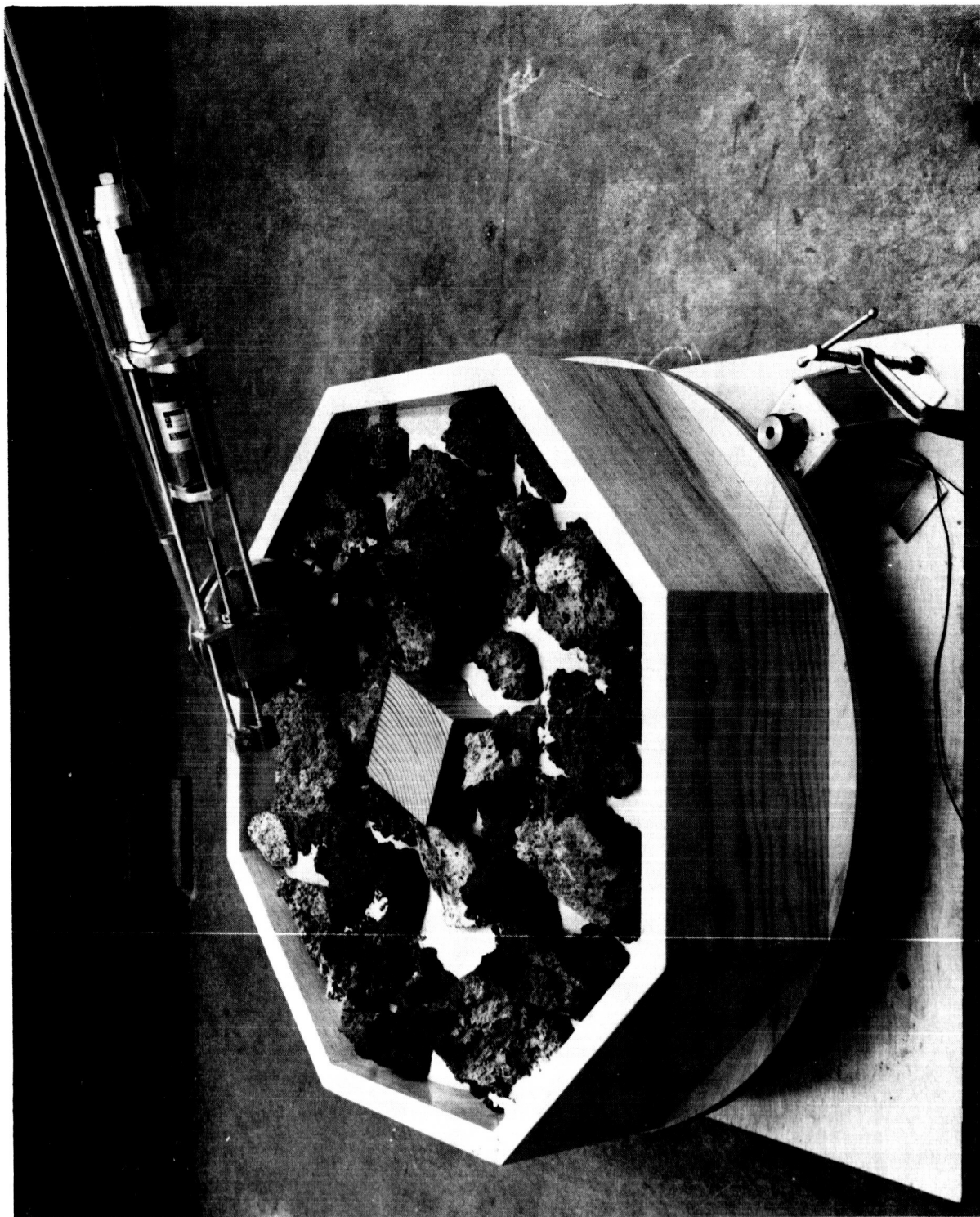


FIGURE 3-5 ROTATING WIRE BRUSH BREADBOARD AND SAMPLE BIN

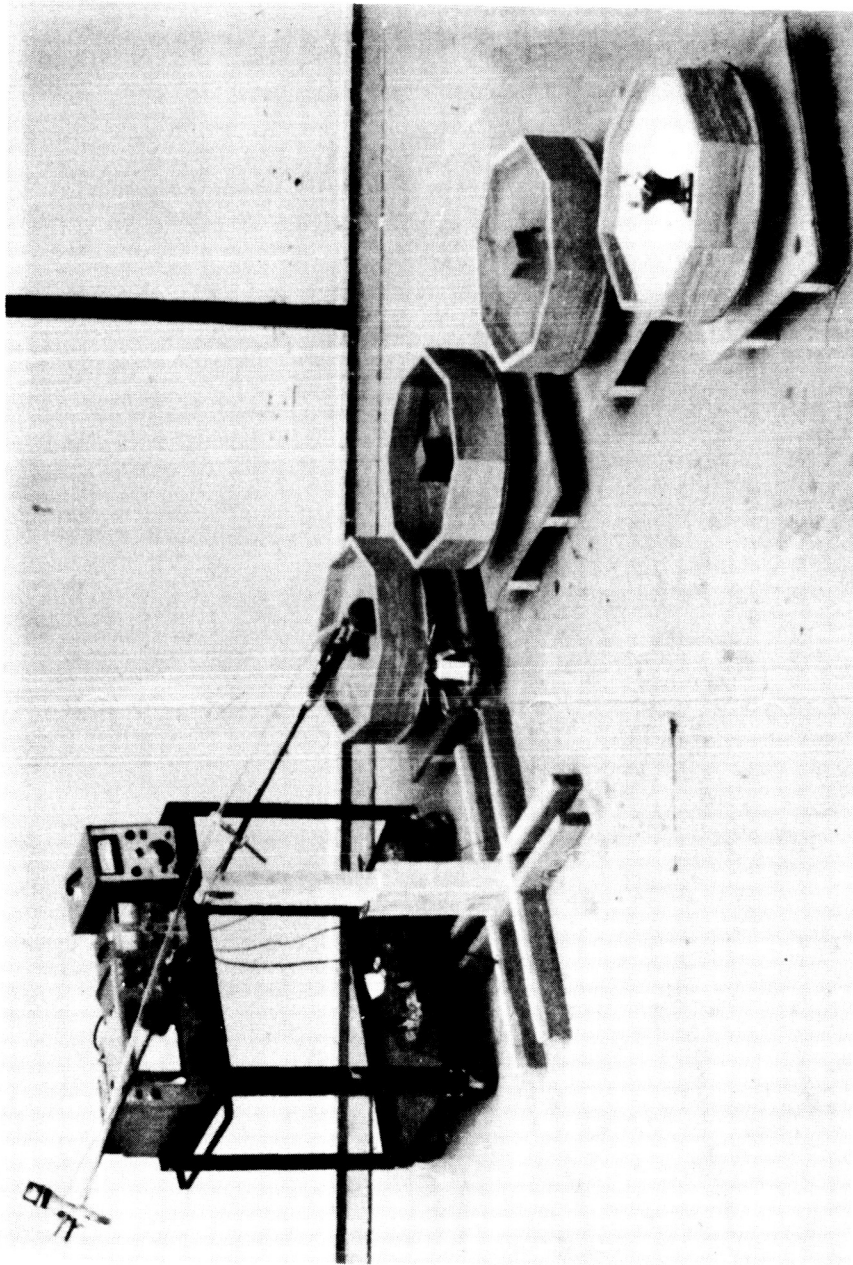


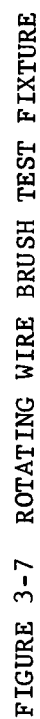
FIGURE 3-6 ROTATING WIRE BRUSH BREADBOARD TEST SETUP

The detail construction of the wire brush breadboard sampling head is shown in Figure 3-7 (PD25185). The wire brush is mounted on the end of the fixture between two side plates. These were designed so that various hood configurations could be installed or changed with a minimum of modification to the breadboard. The motor was mounted in a cradle between the support rods. The motor torque was reacted by a torsion bar. The deflection of the torsion bar is directly proportional to the applied torque.

Torsion bars with a length of 4 inches and diameters of 0.045 and 0.062 inches were made and calibrated. The calibration curves are shown in Figure 3-8. The rms curves were obtained from three sets of observed data and are fairly linear in the intended operating range. The displacement to the right of the theoretically predicted value is due to breakaway torque needed to overcome the friction in the support bearing.

Attached to the motor cradle is a wiper arm which slides along a wire wound potentiometer. The output of this potentiometer as well as the input current were recorded continuously by a 2 channel Sanborn recorder. The voltage was held constant by the power supply and was recorded at the start of each run. The rotational speed of the brush was controlled by the applied voltage. A calibration curve of brush rotational speed is shown in Figure 3-9. This curve was obtained by running the breadboard test fixture completely assembled but not in contact with the surface being sampled. A similar rotational speed calibration curve was obtained for the rotating soil sample bins as shown in Figure 3-10. In both cases, it is seen that the rotational speed varied linearly with the applied voltage. Since the surface speed of the soil model is a function of the radial distance from the center, a cross plot was made relating voltage and radial distance required to produce a surface speed of one foot per minute. This is shown in Figure 3-11. This curve was used to adjust the sample bin speed as required to produce one foot per minute surface velocity traverse.

Before the breadboard testing was initiated, a simpler set up in which a wire brush was mounted directly on a motor shaft was fabricated in order to observe the action of the soil particles abraded by the brush. With no shroud or hood it was observed that the soil was thrown up ahead of the brush in what appears to be a pattern of composite ballistic trajectories. Theoretical vacuum trajectories were calculated and plotted for several brush speeds. These are shown in Figures 3-12, 3-13, and 3-14 for rotational brush speeds of 300, 200, and 100 rpm respectively. The rotational speed of the preliminary breadboard set up was 80 rpm and produced a pattern very similar in appearance as the theoretical trajectories shown in Figure 3-14 except that the maximum height was in the order of 1 inch. These observational tests were made by pulling the brush slowly through about a quarter of inch of basaltic silt with a mean particle size of 32 microns. A simple hood was fabricated for this sampler as shown schematically in Figure 3-15.



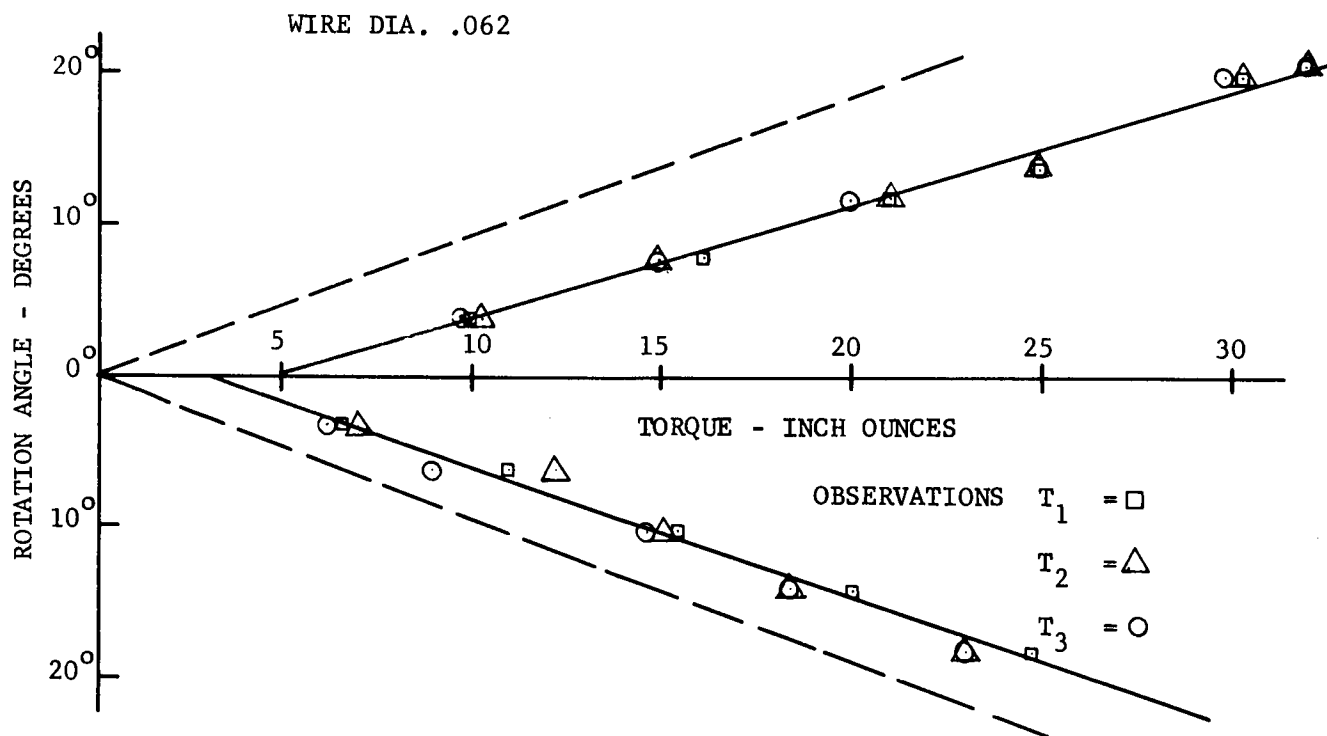
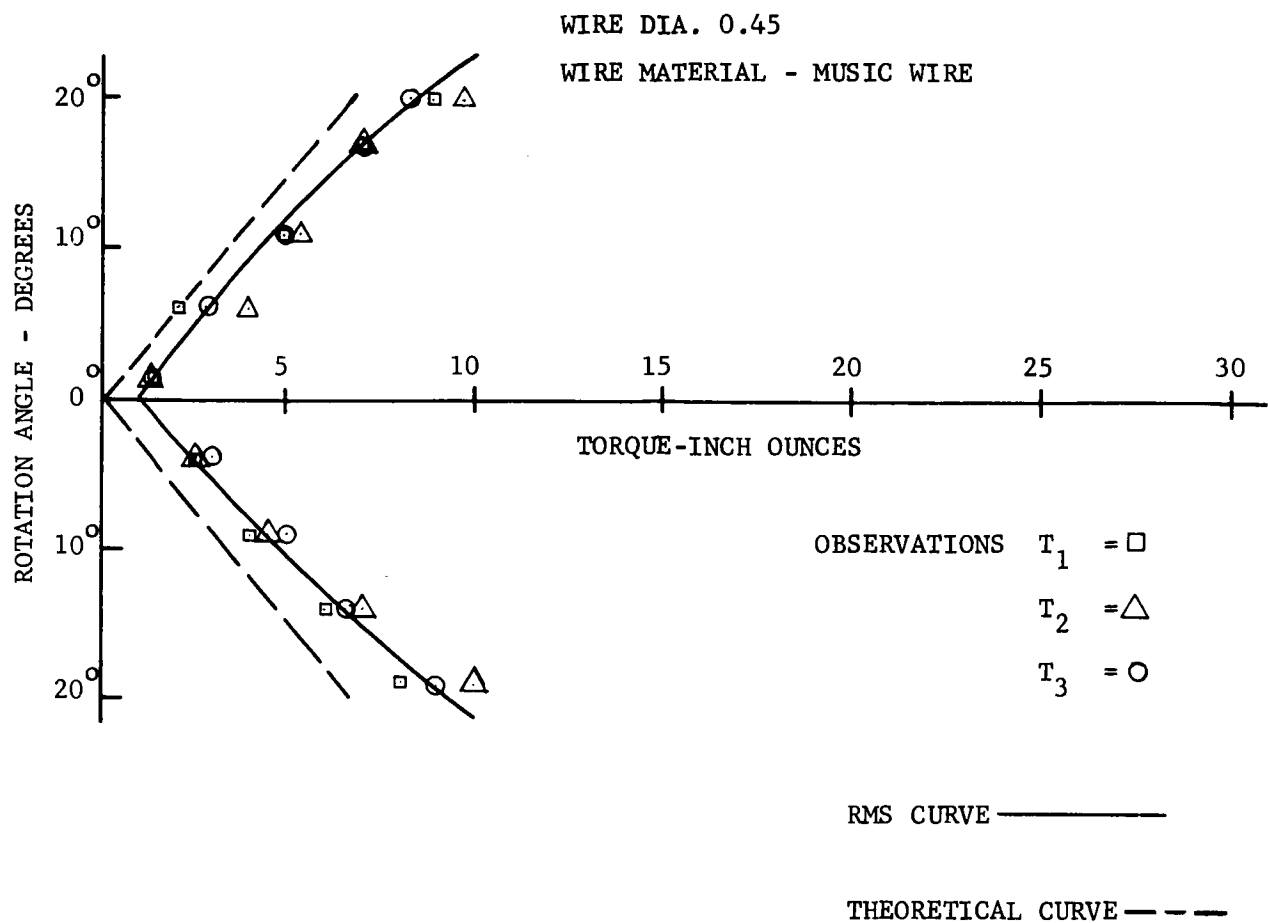


FIGURE 3-8 TORSION BAR CALIBRATION CURVES

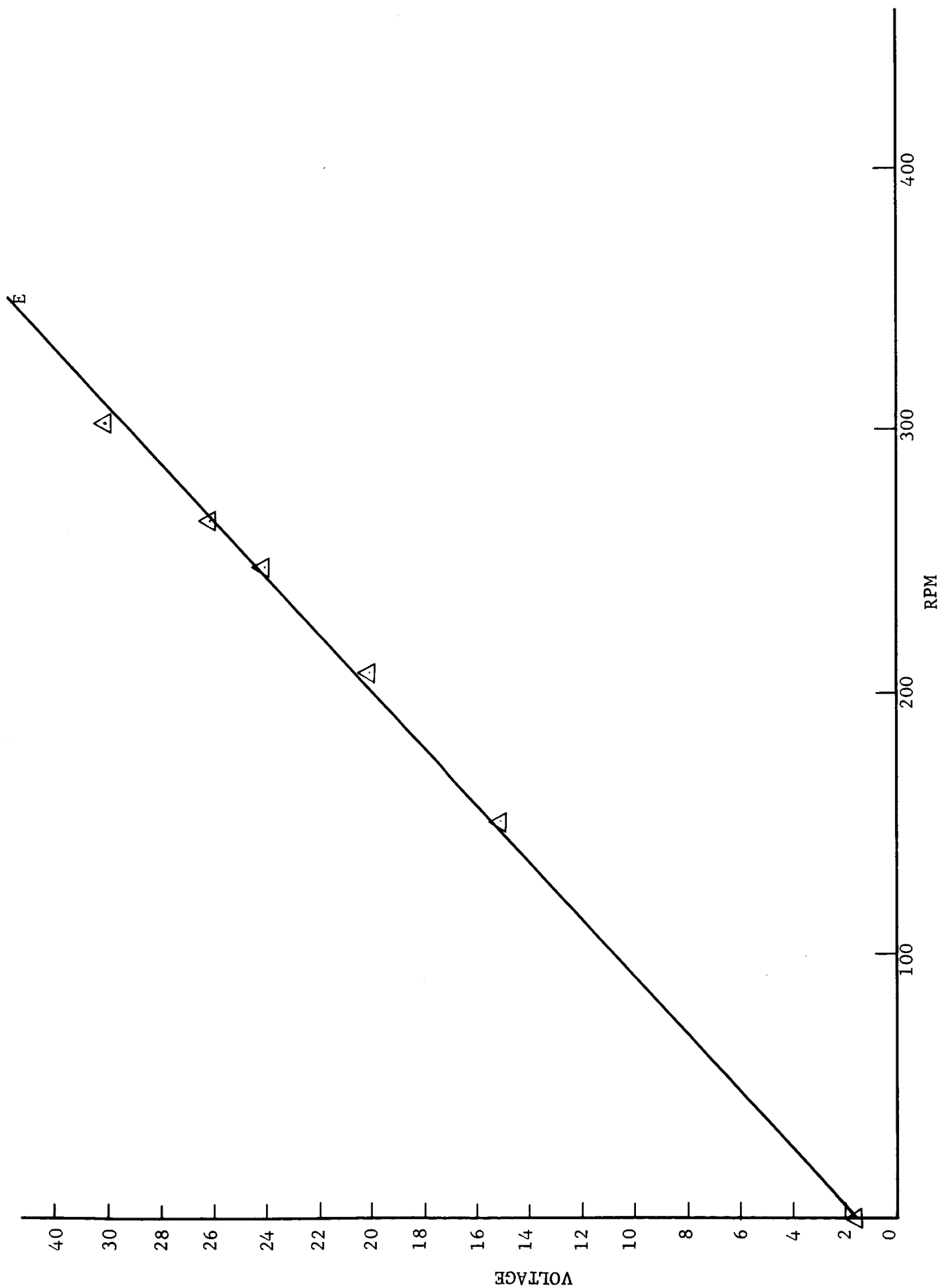


FIGURE 3-9 ROTARY WIRE BRUSH RPM VERSUS VOLTAGE ABL SOIL SAMPLER WITH HOOD AND BRUSH

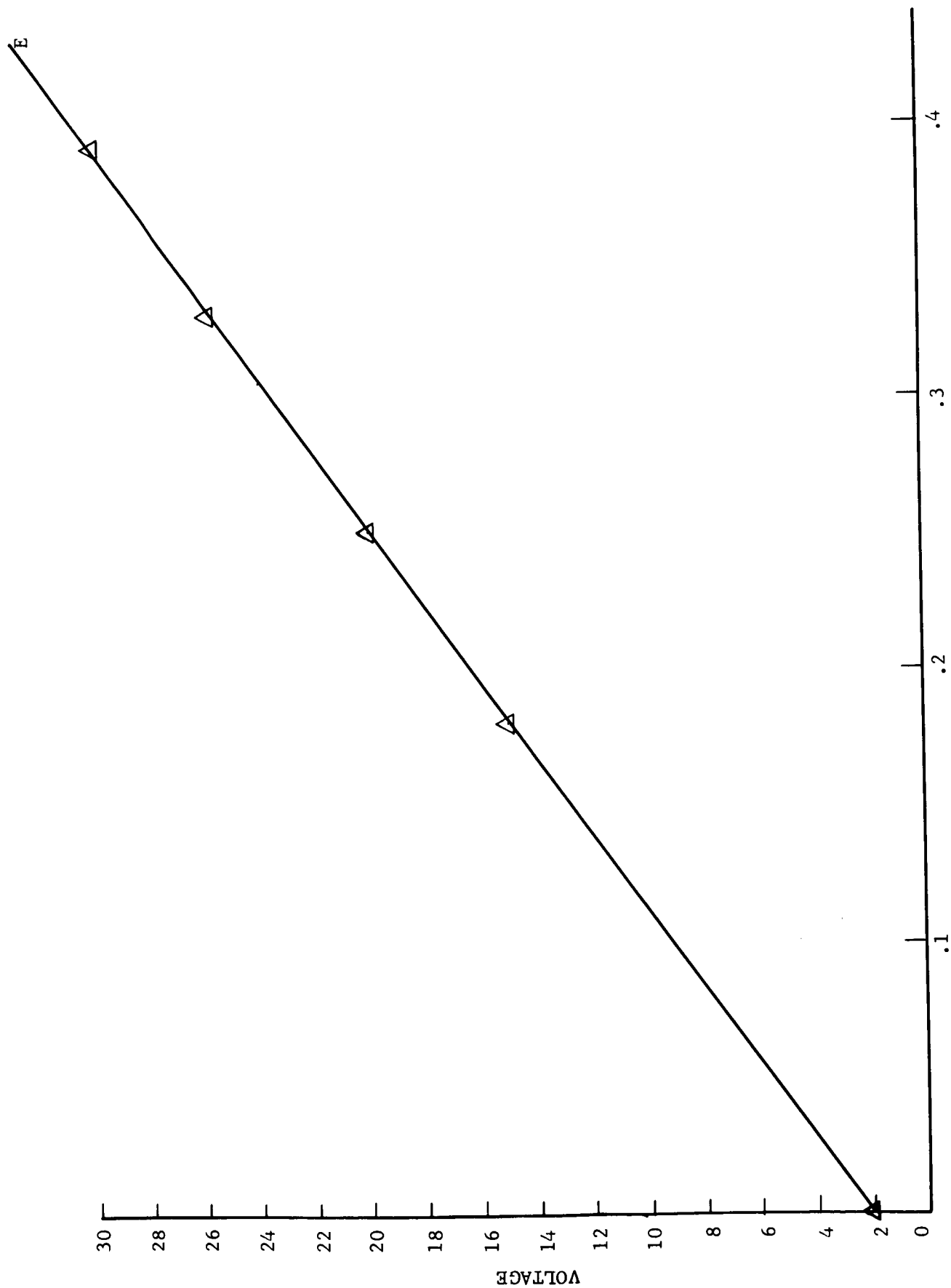


FIGURE 3-10 ROTARY BIN RPM VERSUS VOLTAGE ABL SOIL SAMPLER

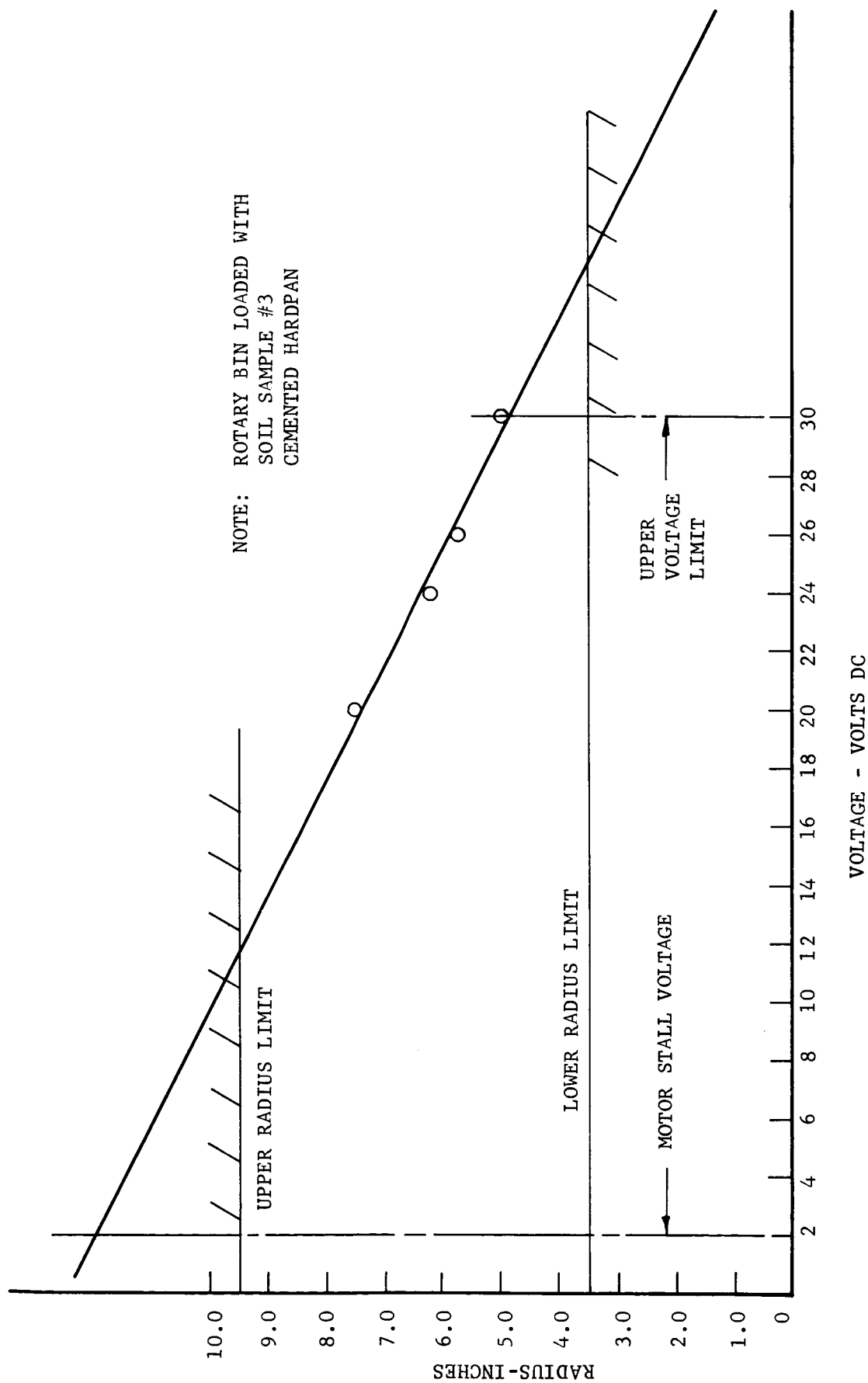


FIGURE 3-11 BIN ROTATION REQUIRED VERSUS RADIAL DISTANCE

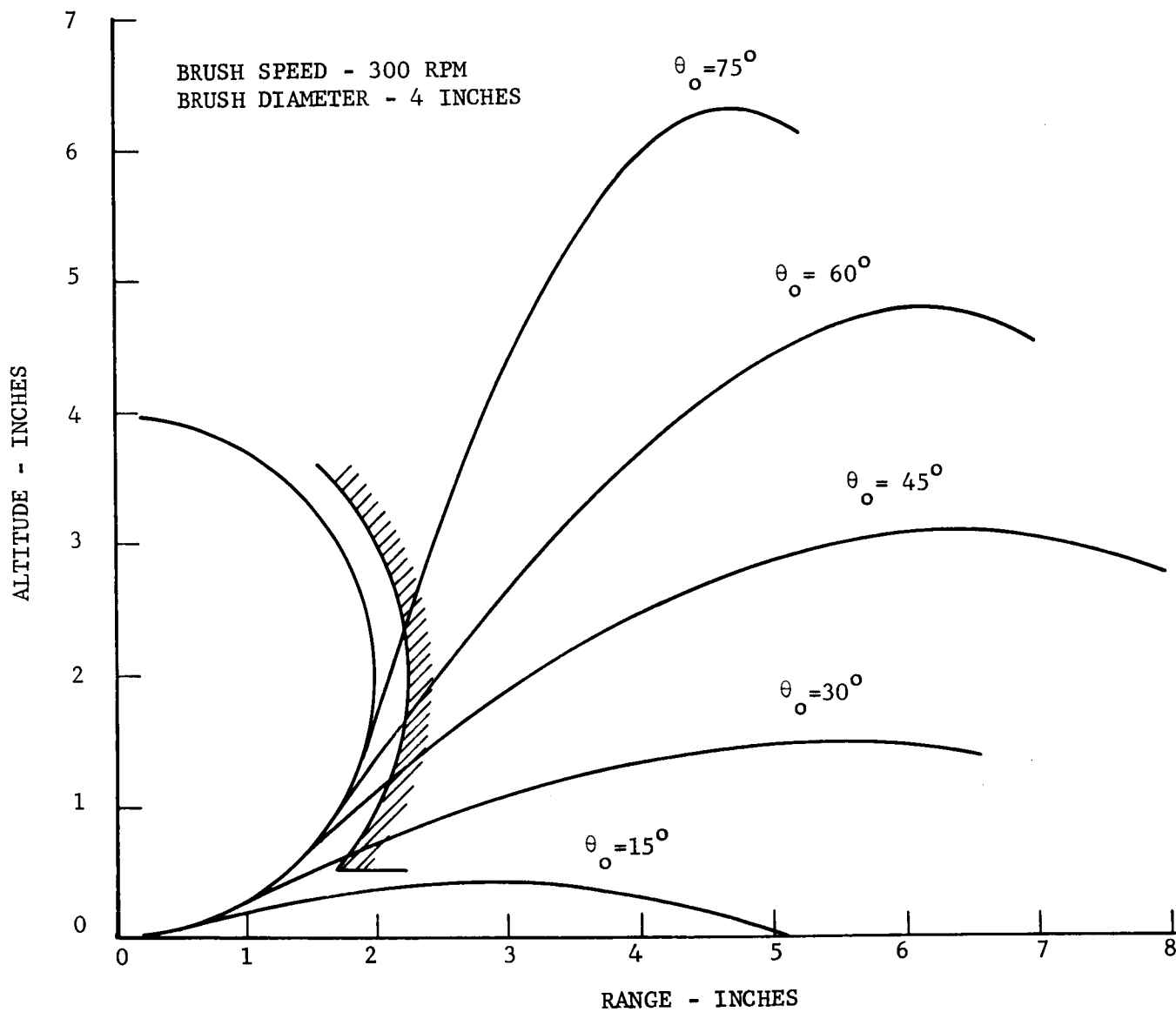


FIGURE 3-12 WIRE BRUSH SOIL PARTICLE TRAJECTORIES

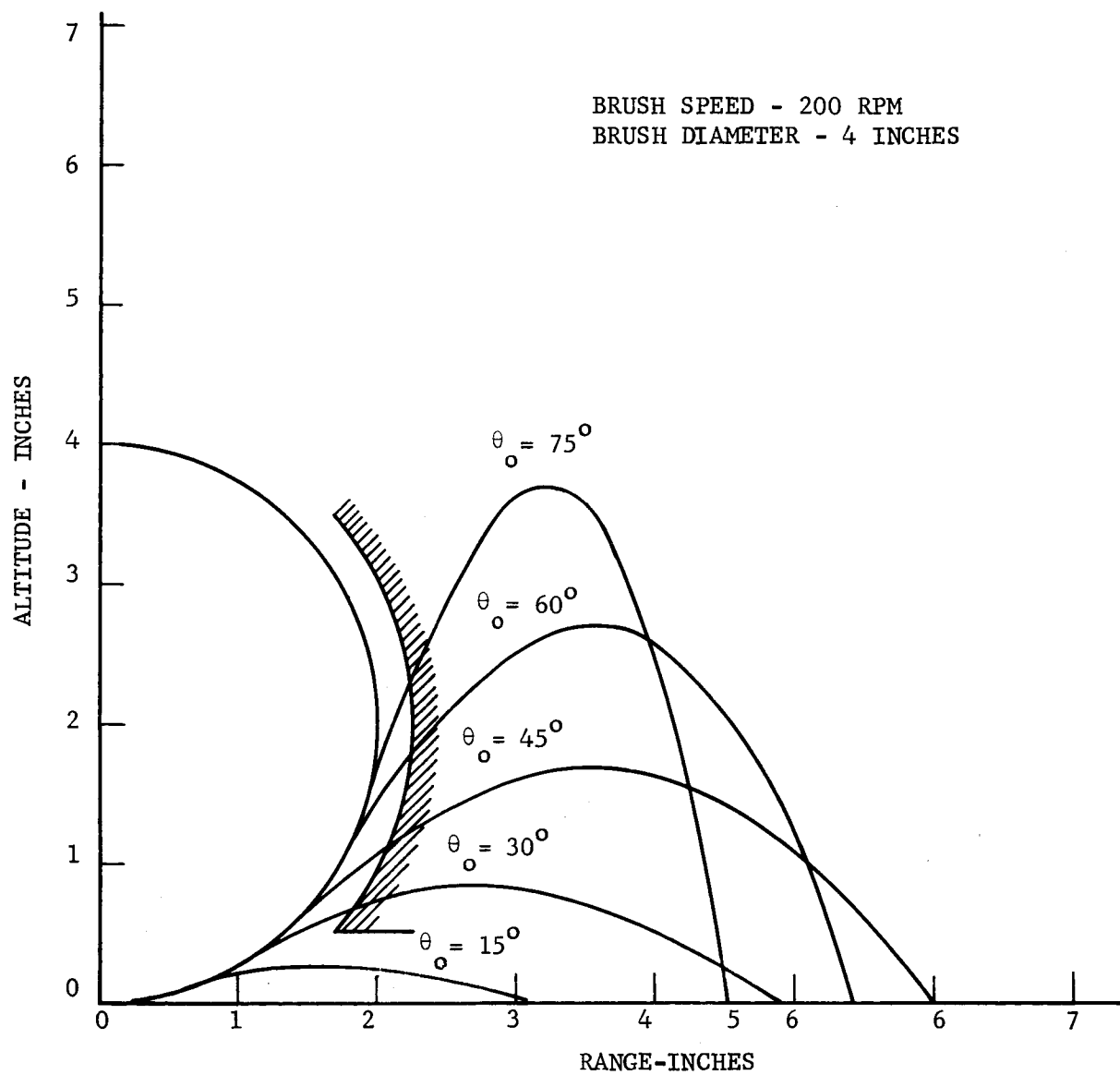


FIGURE 3-13 WIRE BRUSH SOIL PARTICLE TRAJECTORIES

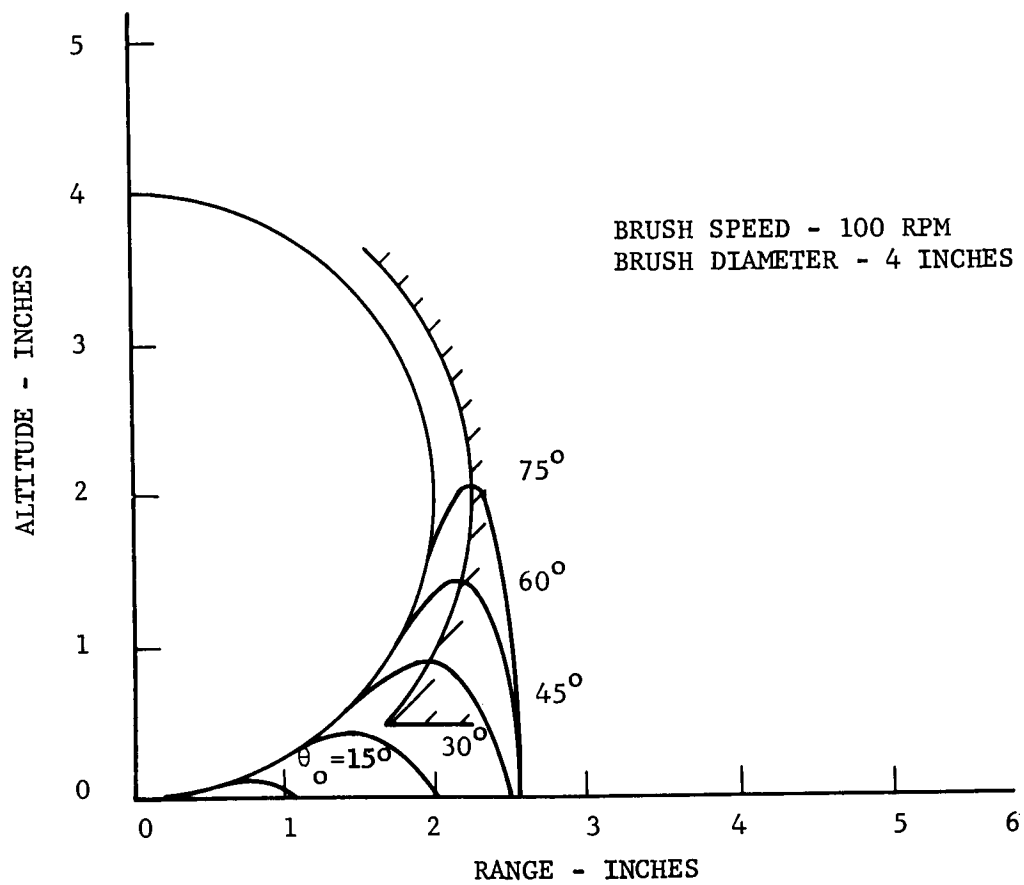


FIGURE 3-14 WIRE BRUSH SOIL PARTICLE TRAJECTORIES

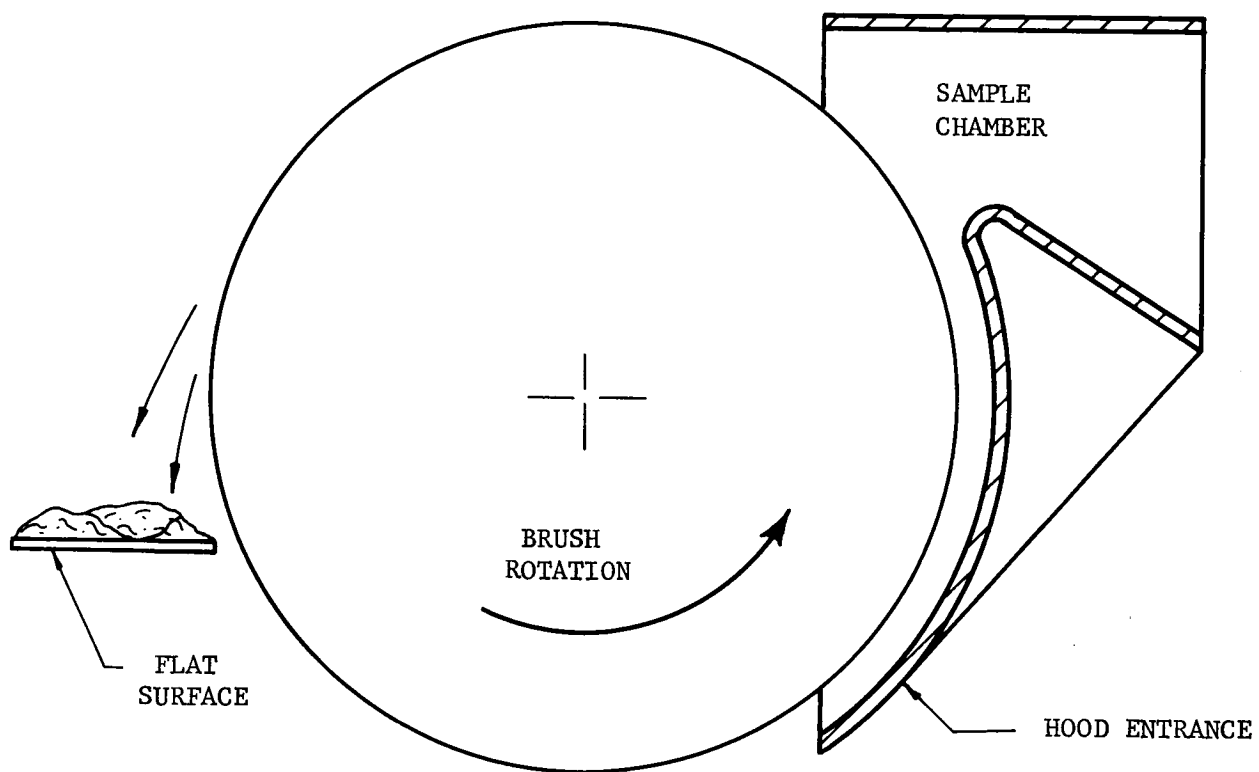


FIGURE 3-15 PRELIMINARY WIRE BRUSH HOOD CONFIGURATION

No sample was collected in the sample chamber with this configuration; however, it was observed that large quantities of soil could be rapidly collected by holding a flat surface on the descending side of the brush as shown in Figure 3-15. The most plausible explanation for this behavior is that gravity forces interact with the impact and centrifugal forces caused by the wire brush carry the soil on around the brush. The following sequence of events are believed to occur.

- (1) Soil is carried up in a variety of ballistic trajectories depending on the point at which it leaves the brush. Soil will pile up in front of brush up to the angle of repose. This soil then acts as a shroud formed at the surface to direct the majority of the soil particles with an initial velocity vector angle of between 30 and 45 degrees.
- (2) As can be seen from the theoretical trajectories in Figure 3-14, the majority of the soil is thrown up by the brush so that it enters the hood. Once it has entered the soil it is carried along by alternately impacting the shroud and the brush. Additional energy is imparted to the soil particles at each brush impact thereby continuing the transport upward.
- (3) In the particular hood configuration shown in Figure 3-15, no particles fell into the sample chamber because they had been carried past the point where their velocity vectors were vertical. In other words, the particles either fell into the brush bristles to be thrown out later or they followed ballistic trajectories in the same direction as the brush rotates.
- (4) Transport continues in this mode until the bristle impact or centrifugal force throws the particles in a trajectory which allows them to clear the brush and collect on the horizontal plate.

In order to determine whether much of the soil could be expected to fall into the bristles of the brush, the centrifugal accelerations were calculated as a function of rotational speed. This variation is shown in Figure 3-16.

For the preliminary breadboard, it is seen that slightly less than one-half a G is developed indicating that most of the soil probably falls into the bristles when on the upper side of the brush. This would indicate that if it is desired to have the sample exit port located at the top of the brush, sufficient rotational speed should be achieved to exceed one G centrifugal acceleration. From the curve in Figure 3-16 a rotational speed in excess of 130 rpm is indicated. Another trend which can be predicted is the effect of brush size. Tangential brush velocity is a

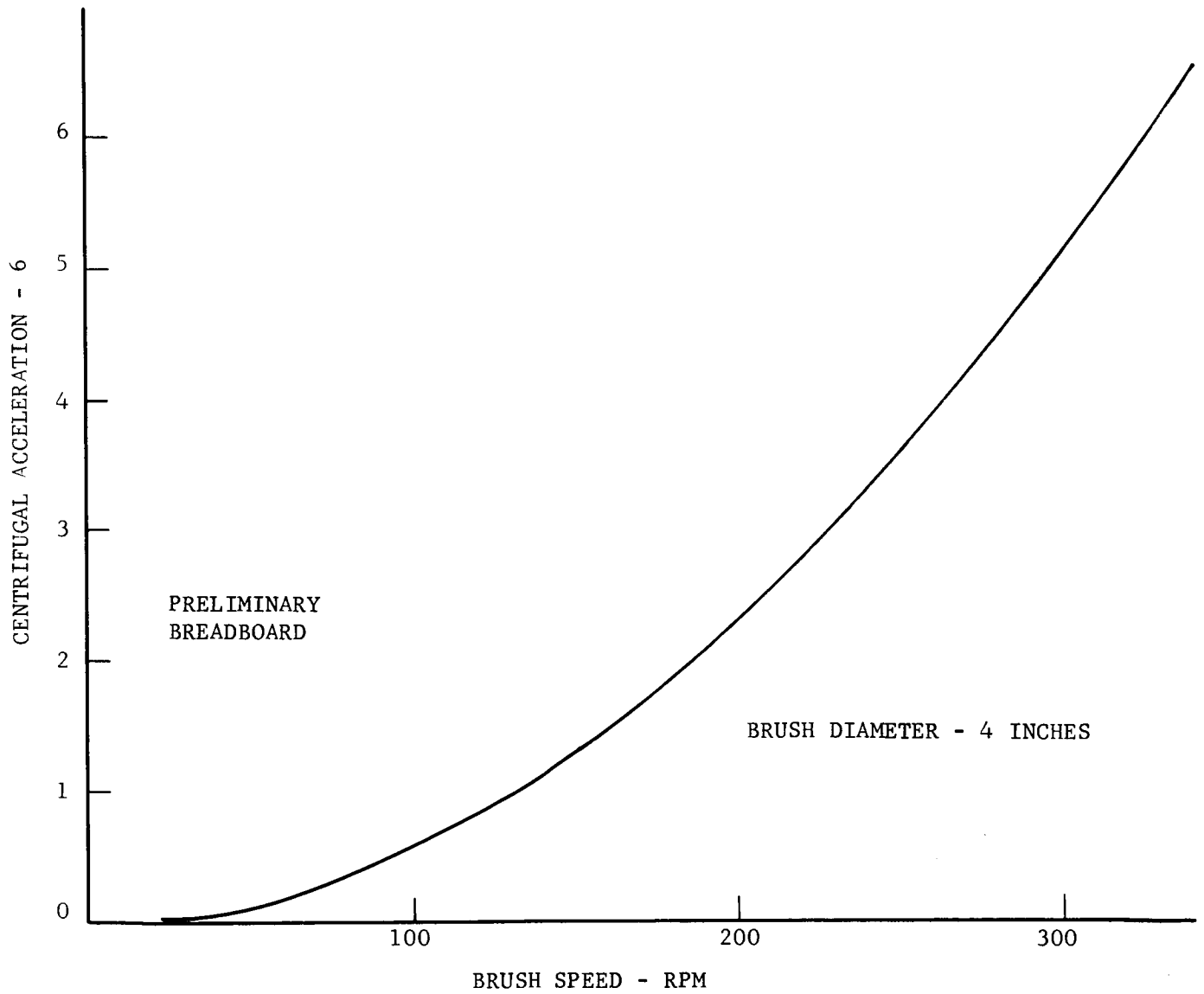


FIGURE 3-16 VARIATION OF CENTRIFUGAL ACCELERATION WITH WIRE BRUSH SPEED

linear function of rotational speed or radius as given by $v = \omega r$. Centrifugal force can be expressed in terms of this tangential velocity by $C_f = Mv^2/r$. Thus, it is seen that for a given tangential velocity, the centrifugal force increases as the brush diameter decreases. This would indicate that the smaller diameter brush should have less of a tendency to load up at a given surface abrasion rate.

Following this preliminary breadboard testing a test matrix was established to guide the efforts in subsequent tests. This matrix is shown in Figure 3-17. In order to maintain control of the relatively large number of variables involved, all initial testing was accomplished on soil model 3 (cemented hardpan). The purpose of the testing as outlined in this matrix was to develop the relative effectiveness of the various parameters affecting the collection of a soil sample by means of abrasion using a rotating wire brush and to obtain quantitative data for use in the design of a soil sampler prototype. The following variables were measured or observed to determine the relative merit of configurational parameters:

- (1) Weight of soil sample collected.
- (2) Soil particle size distribution or bias.
- (3) Electrical input power (watts).
- (4) Mechanical brake power (ft-lbs/sec).

In general, the test philosophy was to reduce the number of variables to be investigated as quickly as possible. Since the hardpan soil model No. 3 is one of the more difficult to collect samples from and, also, the most consistent or uniform in its characteristics, this model was used in all the early testing so that quantitative comparisons of configurational changes could be made meaningfully.

As can be seen from the matrix in Figure 3-17, the first four test configurations are designed to optimize the normal force on the brush and the rotational speed of the brush. The next four test configurations are intended to determine the need for a floating hood and the effect of attack angle on the efficiency of a fixed hood. The next eight test configurations are intended to develop the relative merit of gravity collection versus pneumatic transport and collection. The remaining tests are designed to evaluate the optimized configuration performance in the other soil models. It is noted that the tests shown for soil model No. 5 were tentative subject to prior test results and budget considerations. Soil model No. 5 was a suggested model consisting of a filamentary loaded cohesive soil. This model would provide a severe test in some cases and was therefore felt to be a desirable test model if time permitted. This model was not included in the initial scope of the program.

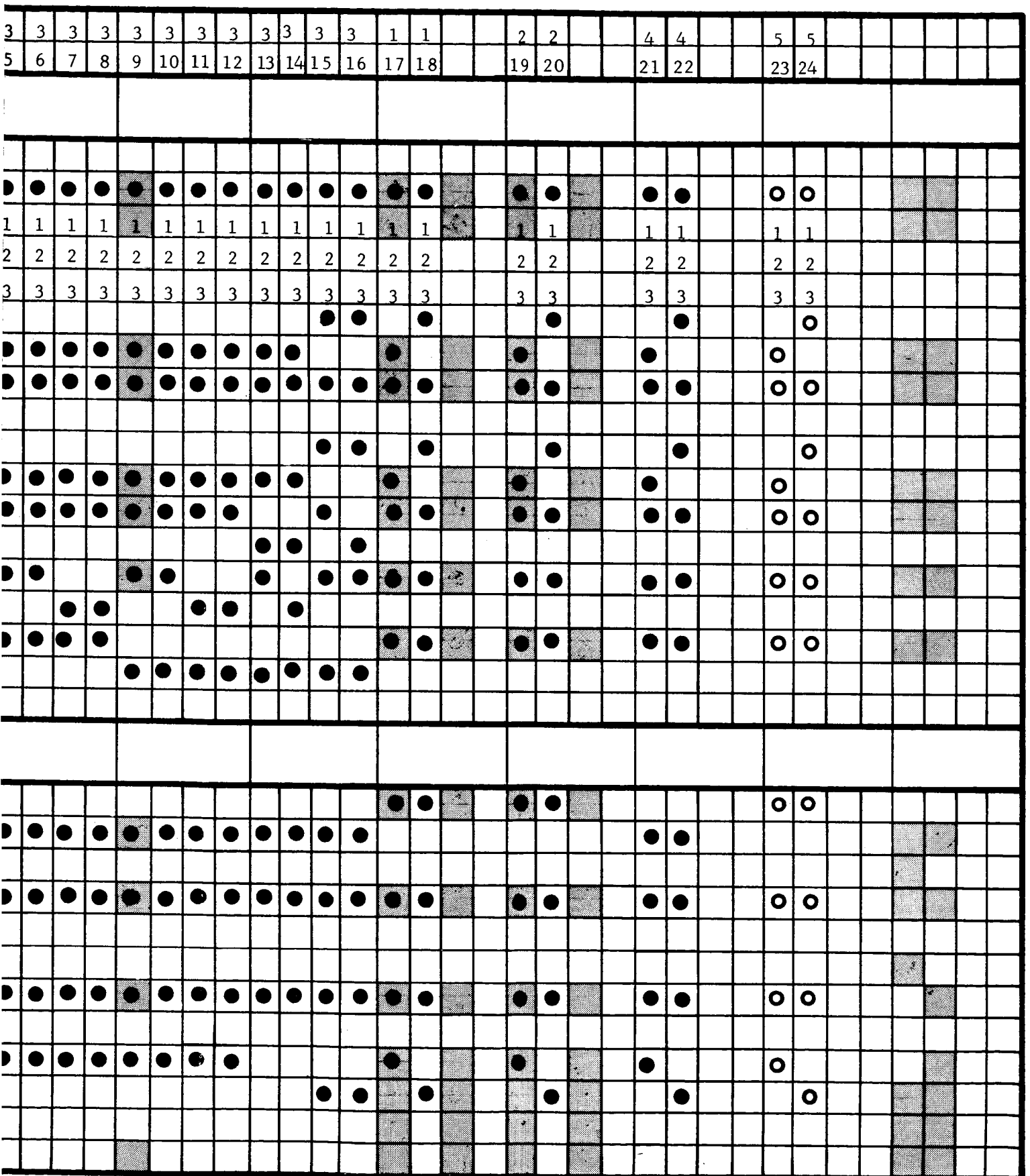
SOIL MODEL		3	3	3	3
TEST CONFIGURATION		1	2	3	4
CONFIGURATION PARAMETERS					
HOOD OR SHROUD TYPE - FLOATING		●	●	●	●
FIXED					
SAMPLING HEAD ATTACK ANGLE - 15°		●	●	●	●
30°		●	●	●	●
45°					
BRUSH DIAMETER - 3 inches					
4 inches		●	●	●	●
BRUSH WIDTH - .25 inches		●	●	●	●
.38 inches					
BRISTLE WIRE DIAMETER - .008 inches					
.010 inches		●	●	●	●
BRISTLE LENGTH - .5 inches w/sideplates		●	●	●	●
1.0 inches w/o side					
BRUSH ROTATION - toward transport tube		●	●		
away from transport tube				●	●
SOIL TRANSPORT MODE - GRAVITY		●	●	●	●
PNEUMATIC					
DYNAMIC PARAMETERS					
TEST DURATION - 1 minute					
5 minutes		●	●	●	●
NORMAL FORCE ON BRUSH - .5 lbs		A	A	A	A
1.0 lbs		B	B	B	B
1.5 lbs		C	C	C	C
SURFACE TRAVERSE RATE - 0 ft/min		●		●	
1 ft/min			●		●
BRUSH ROTATION SPEED - 50 rpm		1	1	1	1
100 rpm		2	2	2	2
150 rpm		3	3	3	3
200 rpm		4	4	4	4
300 rpm		5	5	5	5

SOIL MODEL 1 - COMPACTED SAND
 SOIL MODEL 2 - COHESIVE POWDER

3-27-1

SOIL MODEL 3 - CEMENTED HARDP
 SOIL MODEL 4 - RUBBLE
 SOIL MODEL 5 - FILAMENTARY

FIGURE 3-17
 WIRE BRUSH TEST MATRIX
 CORROSION EVALUATION



AN

The test apparatus consists of a breadboard model mounted on the end of a pivoted boom (shown in Figure 3-6). The rotating wire brush is mounted inside a collection hood which may be either fixed or floating to accommodate itself to the local surface. The brush is driven through a gear train by a motor mounted in a cradle. The torque exerted on the cradle is reacted by a torsion bar spring which allows the cradle to deflect in proportion to the torque. A wire wound potentiometer is mounted to the boom allowing a contact wiper to sweep from the null position thus producing a voltage proportional to the torque. This voltage and the input current to the motor are recorded continuously on a two-channel oscillograph.

In the gravity collection model a plastic container is mounted on the collection hood to collect the soil sample carried up by the brush as gravity causes it to fall toward the surface. In the pneumatic transport and collection mode, a plastic tube is connected to the hood outlet and to a cyclone collector mounted at the other end of the pivoted boom. Counterweights can be applied to the end of the boom to adjust the normal force acting on the brush to any desired value.

In this sampling concept, it is intended that the soil sampler will be mounted on the end of an extendable boom which is retracted continuously during sampling thereby linearly traversing the soil sample head over the surface. To simulate this in the testing, the soil models were prepared in circular bins or trays which are supported on rollers. The bin is driven by a motor causing it to rotate under the soil sampling head, thus simulating the traverse to be encountered in the prototype design and operation.

The following is the general test procedure that was followed until test results indicated that the procedure should be revised.

- (1) Recondition the soil model in the tray to a uniform starting configuration as defined for the particular soil model. The sand and cohesive powder models will be raked smooth and compacted by vibration. The hardpan model is resurfaced by abrasion to remove all tracks of previous runs until it becomes so thin that it must be recast.
- (2) Check configuration conformity with that required by the test matrix and record with a photograph.
- (3) Adjust balance to provide the desired normal force on the abrading head.
- (4) Set motor voltage to produce desired rpm of the wire brush.
- (5) Start the wire brush motor and soil bin motor. During the test run, record (a) elapsed time, (b) voltage and current input, and

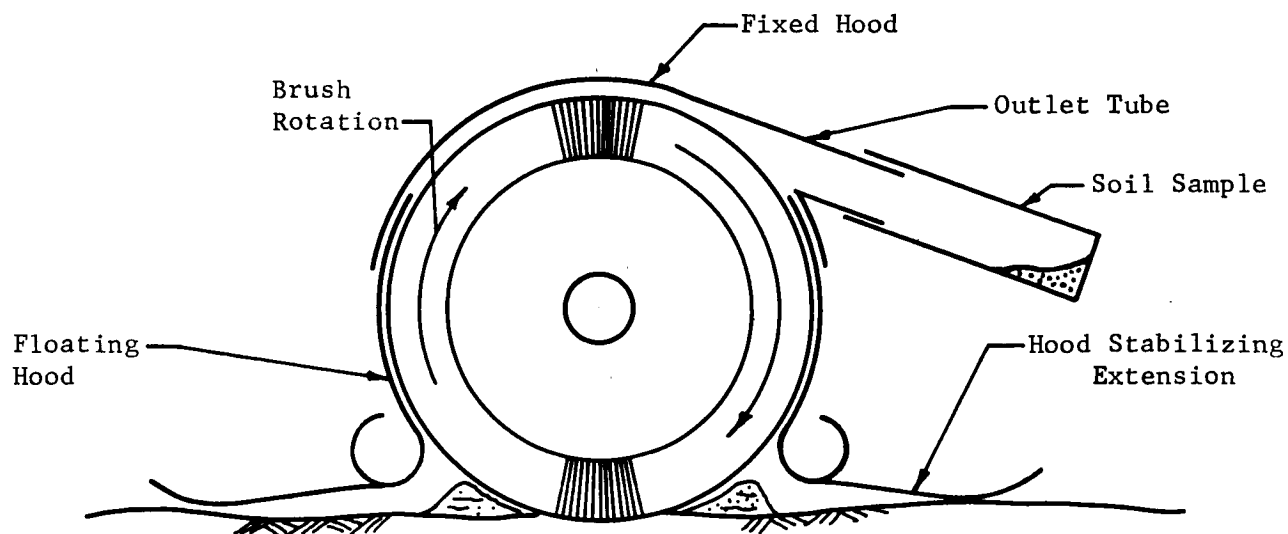
- (c) reaction torque. Observe and note the dynamic response characteristics of the head such as stalling, bouncing, burial or choking of the collection system.
- (6) At the end of the run stop the motors and record the test set-up with another photograph before moving or disturbing the set-up.
- (7) Remove the soil sample collected and preserve for subsequent weighing and analysis of soil particle size distribution.

The blocks in the test matrix shown in Figure 3-17 which are shaded indicate those test configurations which were actually performed.

Forty-five test runs on test configurations 1 and 2 were completed. These utilized a floating hood with mechanical soil collection only, as shown schematically in Figure 3-18. The traverse rate over the soil was zero for configuration 1 and one-foot-per-minute for configuration 2. These tests are designed to evaluate the effect of normal force on the brush, rotational speed of the brush, and traverse rate.

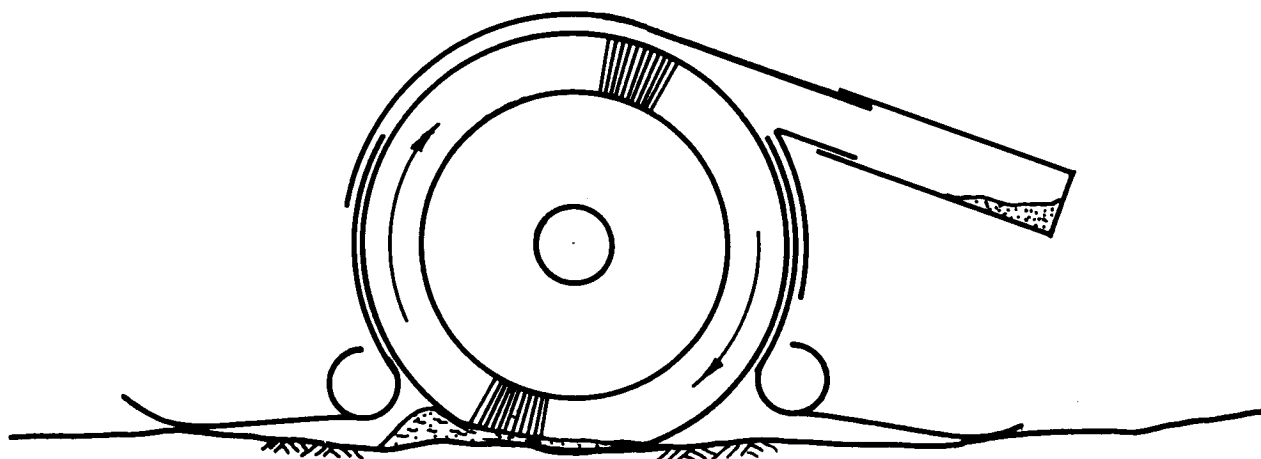
The effect of brush rotational speed is shown in Figure 3-19. No measurable soil sample was collected until a speed of 150 to 200 revolutions per minute was reached. The lines indicated as 1G and 5G correspond to rotational speeds required to produce the centrifugal accelerations as noted. It is interesting to note that soil collection did not begin until 1G level had been exceeded. The explanation for this suggests that the soil particles fall into the brush on the upper side when the centrifugal acceleration is less than 1G and are not thrown into the outlet tube. This is partly substantiated in the tests at zero traverse rate in that a pile of abraded soil sample collected at the point of brush entry into the soil as well as at the point of brush exit. It also suggests that high brush speeds will be more efficient in throwing the soil particles across the gap into the outlet tube. These results confirm the trends inferred in the preliminary breadboard testing.

The data from which Figure 3-19 was obtained was cross plotted as shown in Figure 3-20 for a fixed brush rotational speed of 300 rpm. This is the upper limit achievable with the current breadboard design. In Figure 3-20 it is seen that the traversing sampler is more efficient than the static sampler. This can be expected for two reasons. First, the hood bottoms out on the soil surface when the brush remains in a fixed spot thereby limiting the volume of soil which can be reached by the brush. Secondly, the abraded soil tends to collect at the point where the brush leaves the soil. This mound of loose particles tend to act as a close fitting hood directing the particles into the floating hood. For the traversing sampler, the brush is continually advancing into this mound reworking the loosened soil and rebuilding the mound.



TRAVERSE 0 FT/MIN

CONFIGURATION 1



TRAVERSE 1 FT/MIN

CONFIGURATION 2

FIGURE 3-18 ROTATING WIRE BRUSH BREADBOARD SCHEMATIC

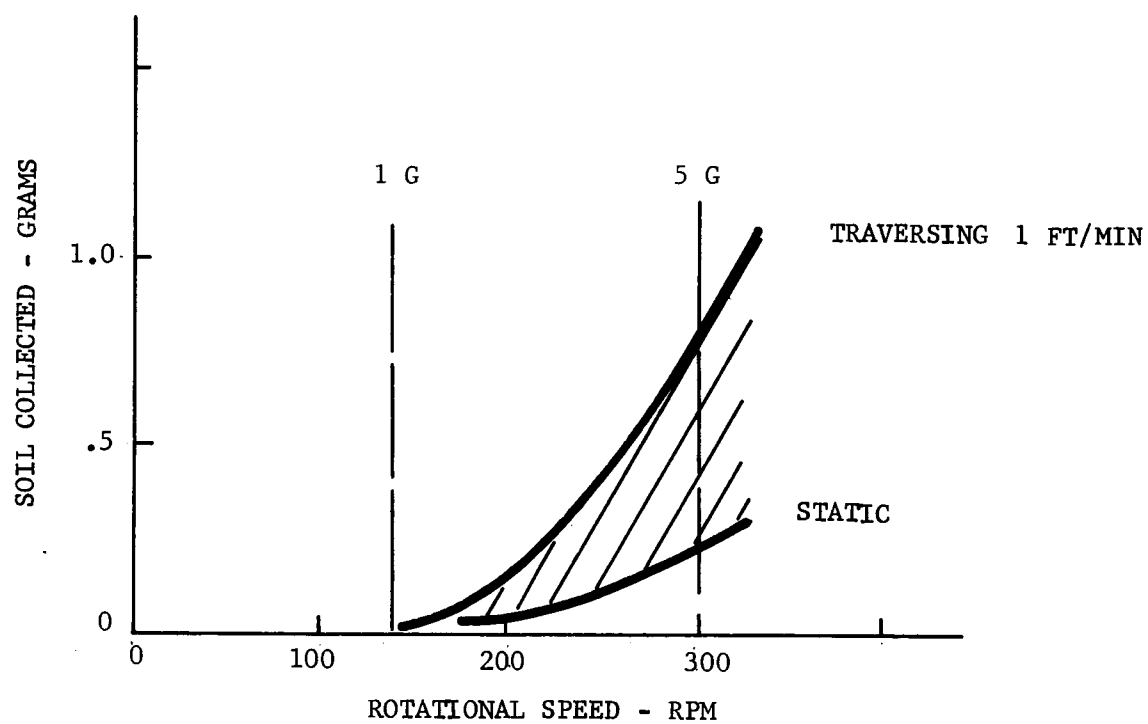


FIGURE 3-19 EFFECT OF BRUSH ROTATION SPEED

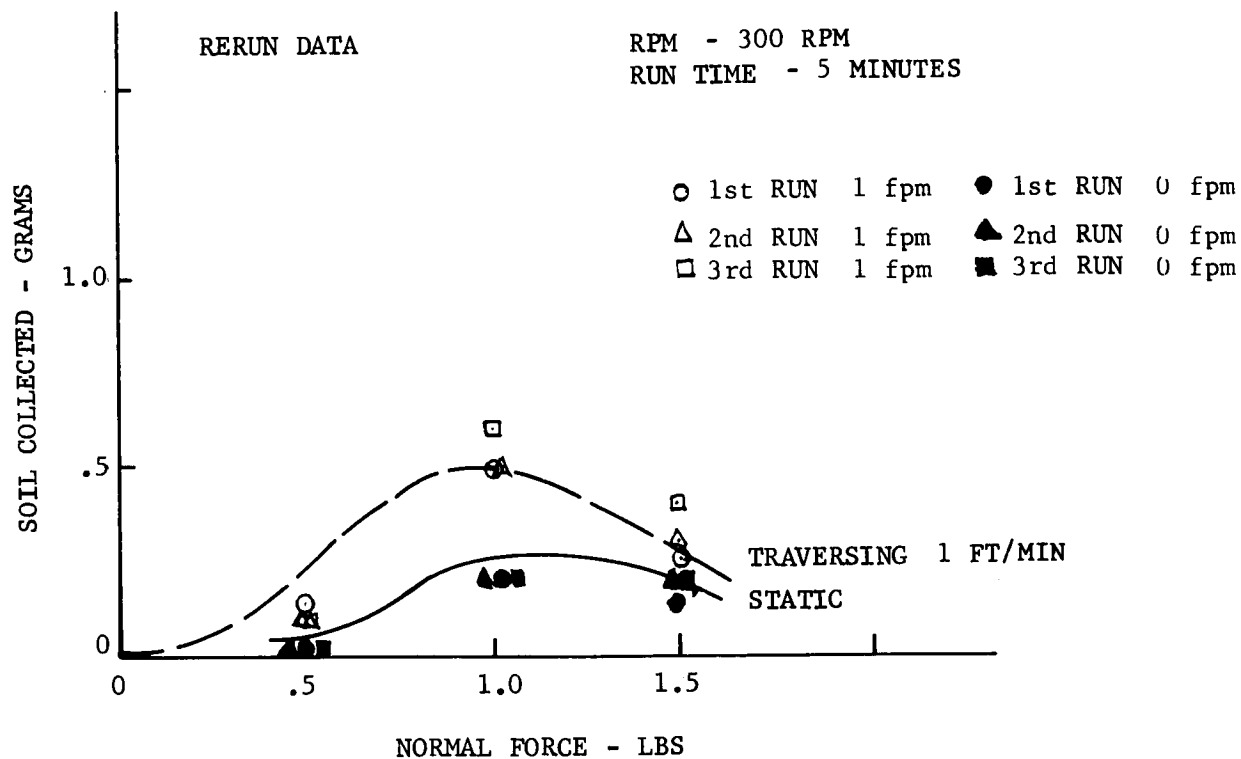
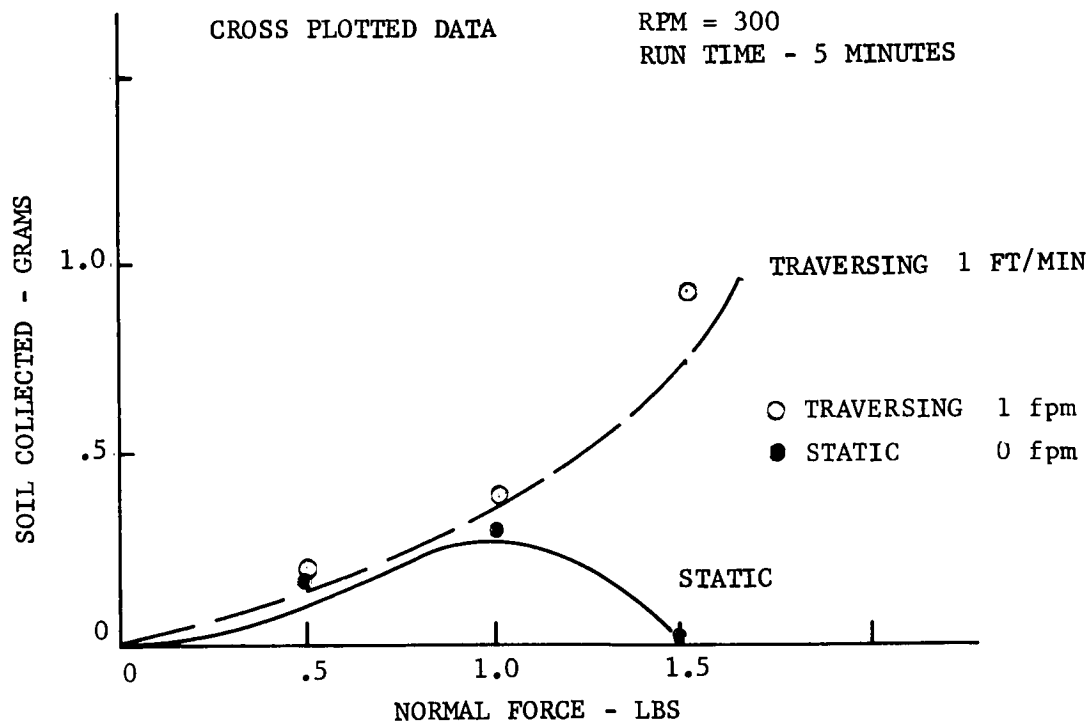


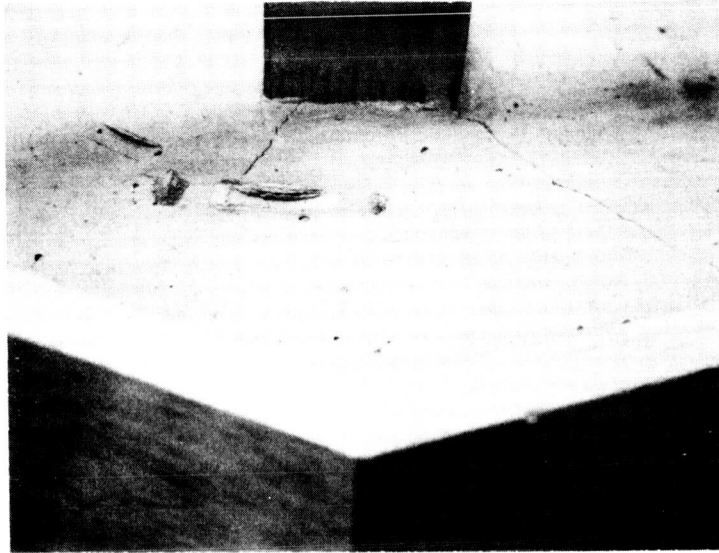
FIGURE 3-20 SOIL COLLECTION AT 300 RPM

Another effect was observed in that the mound achieves a maximum size after which additional soil that is not transported into the hood flows to the side of the wire brush. Referring again to Figure 3-20, it is also seen that an optimum normal force on the brush occurs at one pound. Since initial stalls at 1.5 pounds were observed to be more frequent, this effect is probably due to overloading the motor in torque. The lower set of curves were reruns of the data obtained at 300 rpm to obtain some statistical confidence in the results. The lack of an optimum value for the traversing sampler in the initial data was determined to be caused by faulty test control. The soil bin makes more than one revolution in 5 minutes so that the second time around the brush was traversing the same track it had previously traversed. Since the majority of the soil particles are not collected, most of these are deposited on the track as the brush moves along. Thus, the wire brush was reworking these loose particles on the subsequent traverse thereby enhancing the collection rate. On the reruns, the track was brushed clear after the wire brush had passed to remove this ambiguity from the results.

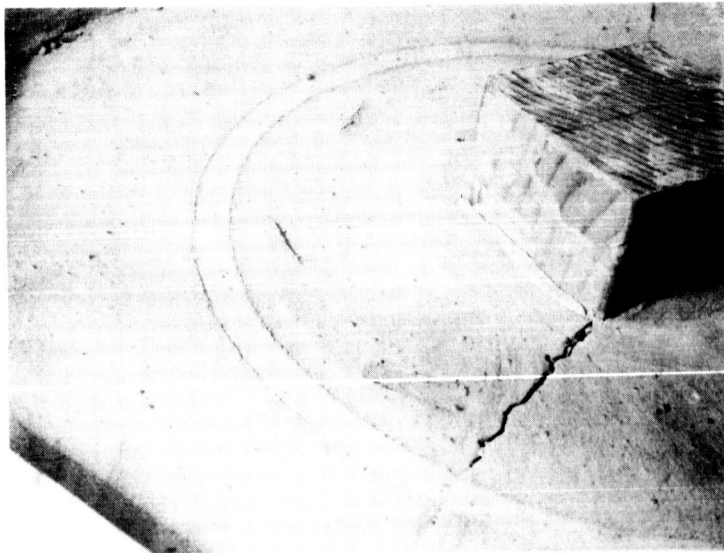
Some typical examples of the abrasion characteristics of the rotating wire brush in the floating hood are shown in the photographs of Figure 3-21. In the case of the static run, with a zero traverse velocity, it is seen that abrasion continued until the wire brush test fixture was resting on the floating hood. It is interesting to note that much larger quantities of soil piled up at the entrance and exit points of the wire brush in the hood than was collected in the sample collection chamber. The sample collected was in the order of .01 to .1 grams depending on brush speed.

It should be pointed out that several characteristics of the floating hood configuration used were not considered to be desirable. One of these was the tendency of the floating hood to ride along with the wire brush due to friction at the shaft. This would cause the hood to hit the ground and lift the wire brush off of the surface. Spring wire extensions prevented this action but rendered the configuration unsuitable for the rubble model. The shape of the sample outlet tube was circular in cross-section where it entered the hood. This presented a variable opening across the width of the hood. It was observed that in the zero traverse rate runs, the piles of soil samples deposited at the entry and exit of the hood were larger at the edges indicating that more soil was transported near the sides of the hood where the opening to the sample collection chamber was minimal.

Based on the observations made in these tests and the ultimate desire to have the simplest possible design, a new fixed hood was designed and fabricated for continued testing. This hood is designed with skirts that allow only 0.06 inches of the brush to extend past the outer radius of the hood over approximately 160 degrees of the total brush diameter.



TRAVERSE VELOCITY, 0 fps



TRAVERSE VELOCITY, 1 fps.

FIGURE 3-21 TYPICAL ABRASION CHARACTERISTICS OF WIRE BRUSH ON HARDPAN SURFACE

Fixed to the edge of the skirts are inlet ports every half inch along the circumference. These are intended to scoop up the soil particles that flow out of the leading mound along the sides of the brush. The side bristles on the brush are trimmed to just clear the inner surface of the inlet scoops so that it will carry the entering soil along to the sample collection ports. This hood configuration is shown schematically in Figure 3-22.

Additional testing was accomplished using the fixed hood configuration of the rotating wire brush breadboard. It was desired to obtain data for the relative brush rotation and surface traverse as shown in Figure 3-23 and referred to as conditions A and B. Since the effect of normal force indicated an optimum value near one pound in previous tests, these test runs were made using this value of normal force. Two soil models were used in these tests. One was the hardpan (Model #3) and the other was a layered model with one-half inch of sand over adobe brick. The soil collection rate achieved in these tests are shown in Figure 3-24 for the hardpan and in Figure 3-25 for the layered model. In both cases it can be seen that condition A represents a more efficient collection configuration than does condition B when only mechanical collection is used. The low collection rate on the hardpan model is augmented by the use of pneumatic transport as was shown in other tests. Mechanical collection only provides adequate collection rates with the layered model. The collected sample for this model was composed not only of sand, but contained some abraded particles from the hardpan underlayment.

During the course of testing with the fixed hood rotating wire brush breadboard, the free running power was observed to rise steadily with use. Prior to running the tests on the layered soil model, the fixed hood was disassembled to be cleaned. On examining the parts it was noted that the spiroid gear was badly worn and wear patterns established the existence of interference or rubbing at the ends of the bristles on the sides of the fixed hood as shown in Figure 3-26. The interferences were relieved by machining some of the metal away in these areas. This suggests that rub strips should be incorporated in the hood design in such a manner that contact can only occur on the sides of the bristles rather than the ends. After relieving the interference areas and replacing the spiroid gear and ball bearings, the free running current was decreased from 1.2 amperes to approximately .3 amperes. This rework was accomplished before the test runs on the layered sand model were made.

Preliminary runs utilizing pneumatic transport indicated that an order of magnitude increase in the quantity of soil sample collected on hardpan could be achieved.

To obtain a check on the effect of pneumatic transport, a few runs in test configuration 11 were made. In this configuration, the wire brush was rotating away from the entrance to the pneumatic transport tube which

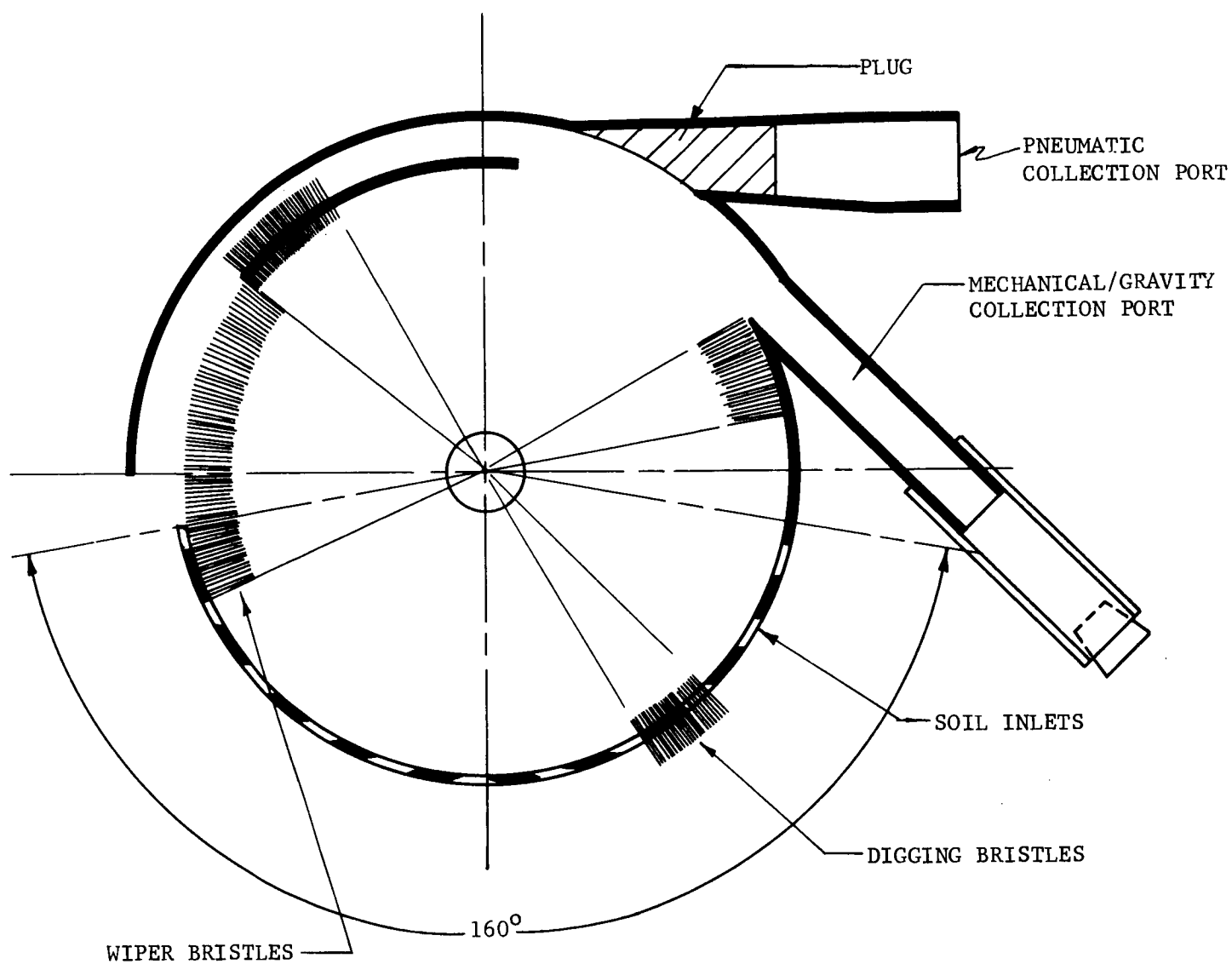


FIGURE 3-22 SCHEMATIC - FIXED HOOD CONFIGURATION

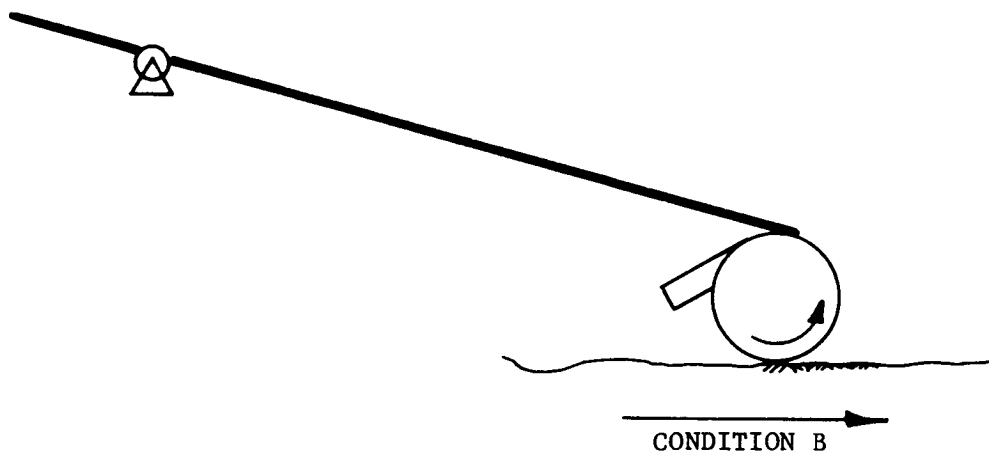
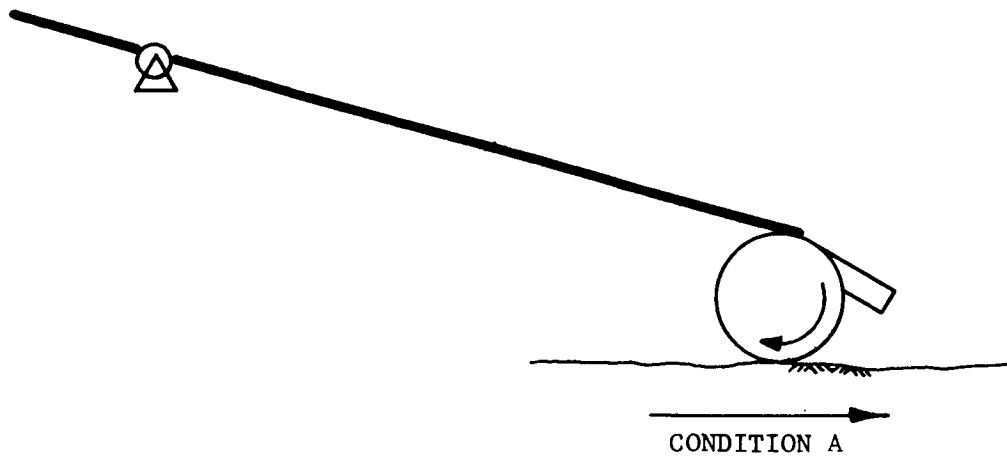


FIGURE 3-23 TEST RUN CONFIGURATIONS FOR MECHANICAL COLLECTION

1 LB NORMAL FORCE
MECHANICAL COLLECTION ONLY

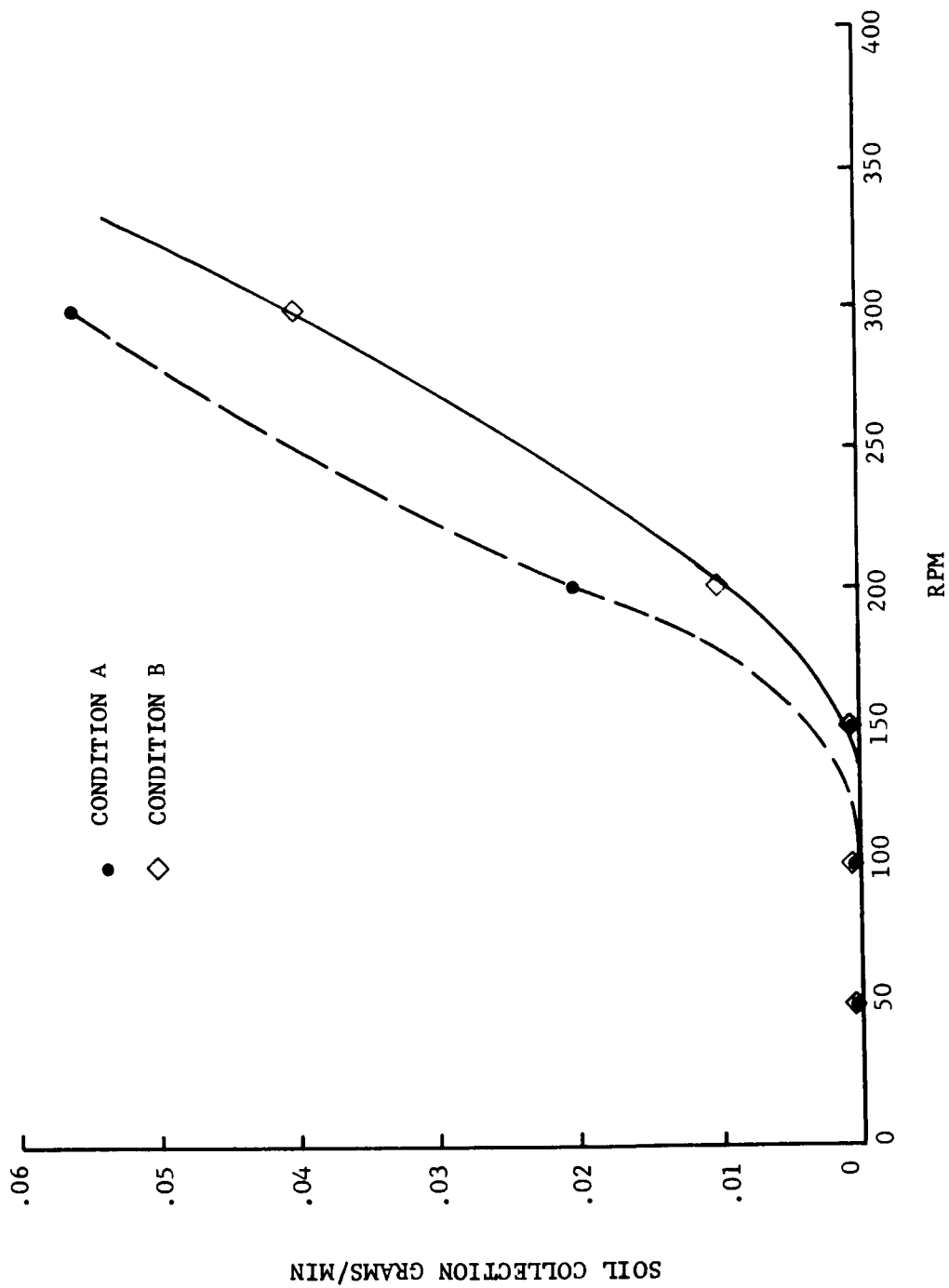


FIGURE 3-24 ROTARY WIRE BRUSH COLLECTION RATE ON HARDPAN

1 LB NORMAL FORCE
MECHANICAL COLLECTION ONLY
5 INCH SAND OVER ADOBE BRICK

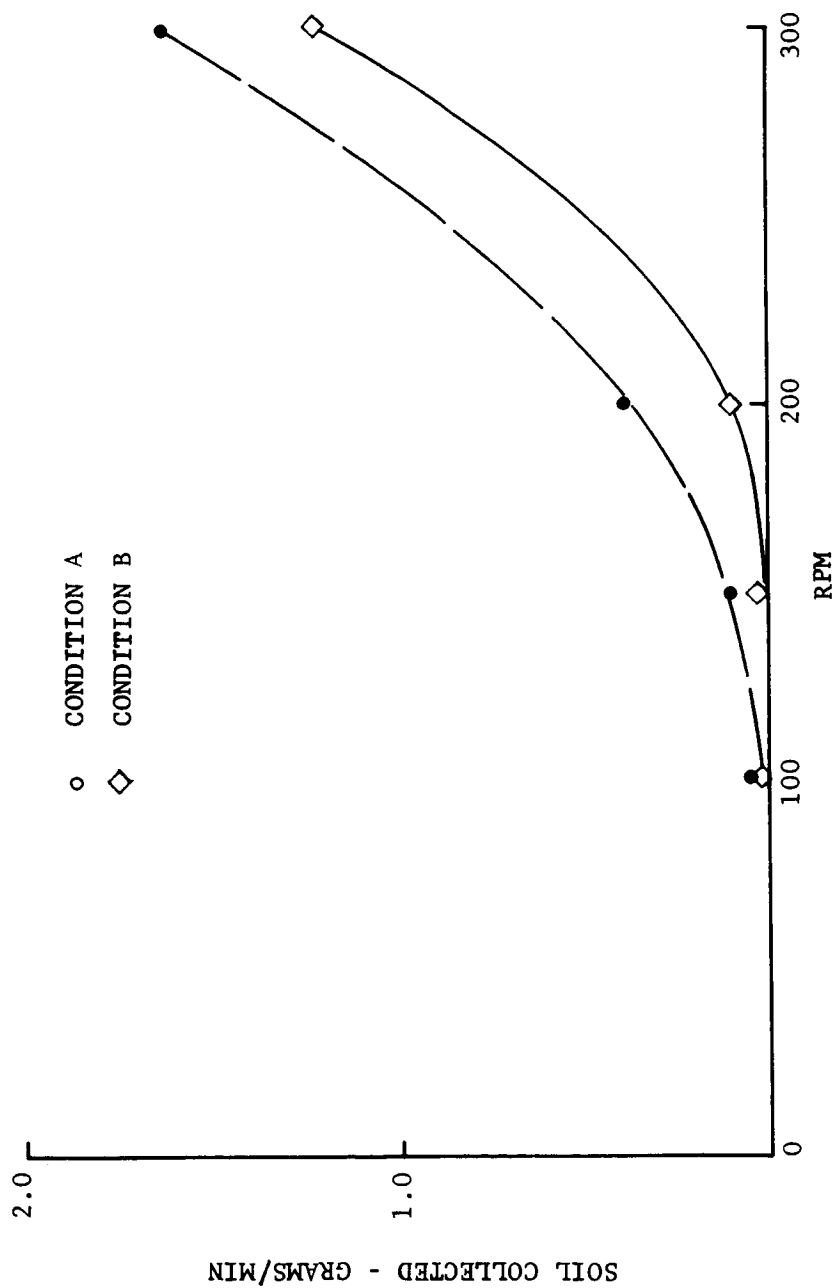
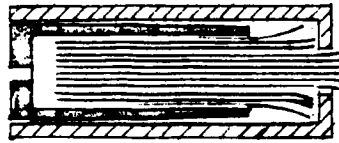


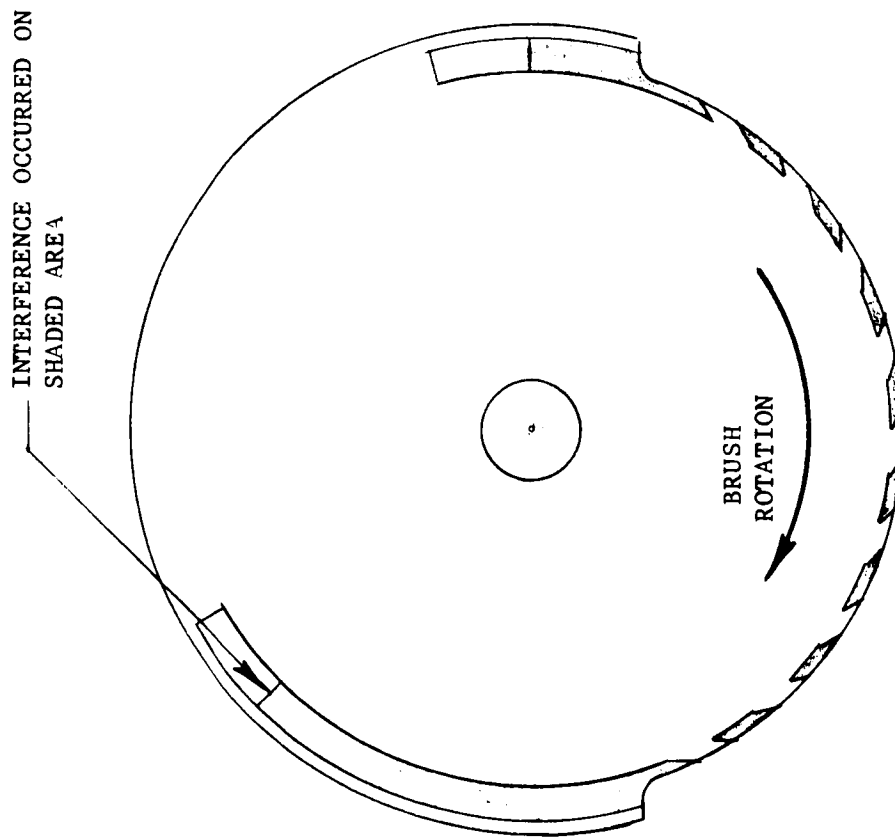
FIGURE 3-25 WIRE BRUSH COLLECTION RATE ON SAND LAYERS OVER HARDPAN

INTERFERENCE OCCURRED ON
SHADED AREA

HEAVY INTERFERENCE
OCCURRED HERE



CROSS-SECTIONAL VIEW THROUGH
HOOD AND BRUSH



INTERNAL VIEW OF
HOOD SIDE PLATE

FIGURE 3-26 WIRE BRUSH BREADBOARD INTERFERENCES

is probably not optimum. Even so, the total sample collected in 5 minutes was approximately 3.5 grams as opposed to 0.5 grams. This represents nearly an order of magnitude improvement in collection rate. No particle size measurements were made on the collected sample; however, the pneumatically collected sample appeared to be very much finer than the mechanically collected sample.

The test configurations referred to in the test matrix of Figure 3-17 as 17 and 17A were run in deep compacted basaltic silt. Test configurations 19 and 19A were run in deep compacted sand. The results were the same for both models. The sample collection chamber was filled in a few seconds after which the sampling head tended to choke up. Runs 17 and 19 were made with soil bin rotating so that the surface was carried towards the sampling head. Due to the attach angle of the boom, the head tended to bury itself into the soil until it stalled. This was not considered to be typical of the proposed prototype sampler mode of operation. Test configurations 17A and 19A were made with the sample bin rotating away from the sampling head and boom. The results were essentially the same except that the sampling head only buried itself in the soil to a depth slightly less than half the diameter of the hood. The results of these tests indicate that some method must be used to limit sampler hood burial in deep loose soils to prevent too much soil from being ingested into the sampler. Alternatively, some form of overflow or dump for the excess material should be provided.

In summary, it is noted that brush rotational speed was maintained as a variable in many more tests than was originally intended and that variations in attack angle did not seem to be too important for the head configurations used. The layered model in test configuration number 26 was added in order to obtain quantitative data for a shallow sandy surface to compare with the results obtained on hardpan. As seen in Figures 3-24 and 3-25, up to two orders of magnitude increase in soil collection can be achieved by mechanical means if the surface is covered with a loose or weak material. It should also be noted that the synthetic hardpan model used in these tests is actually as strong as soft stone and is much harder than natural hardpan.

3.3 PNEUMATIC TRANSPORT BREADBOARD

The initial design study instituted for determining the effectiveness of blower-induced pneumatic tubes as particulate sample transport subsystems consisted of the calculation at the flow velocity and pressure drop in a tube of constant diameter relative to the tube diameter, its length and the atmospheric model utilized. The two JPL Mars model atmospheres used, VM-4 and VM-7, give the extreme boundary conditions. All of the other atmospheres, VM-8, VM-1, VM-2, and VM-3, fall between these two in terms of flow characteristics. The VM-4 atmosphere has a surface pressure of 10 millibars and a surface temperature of 200 degrees K. It is composed of 68 percent carbon dioxide and 32 percent argon by volume. The VM-7

atmosphere has a surface pressure of 5 millibars and a surface temperature of 275 degrees K. It is composed of 20 percent carbon dioxide and 80 percent nitrogen by volume. For comparison purposes, the flow characteristics in a tube for a standard earth atmosphere were calculated. The results of these calculations are shown in Figures 3-27, 3-28, and 3-29. Figures 3-27 and 3-28 are for the Martian atmospheres at two different assumed values of inlet Mach number of 0.05 and 0.10, respectively. Figure 3-29 is for earth atmosphere with an inlet Mach number of 0.05. The data were calculated on the basis of an isentropic expansion for compressible flow which is conservative from the standpoint of predicting the required pressure drop to produce the flow. The circled points on the ends of the curves for the small tubes represent the critical length beyond which an increase in length of the tube causes a change in the inlet flow conditions. From these curves, it is seen that tube diameters of less than 0.5 inch for a 10 foot tube induce penalties due to excessive pressure drops and under certain conditions can choke the flow. Thus the trend is to use larger diameter tubes at greater deployed lengths.

To evaluate the design requirements for pneumatic transport within the telescoping boom, test hardware for the pneumatic components were procured and/or fabricated as illustrated by Figure 3-30. Two transparent acrylic plastic tubes were made with the dimensions shown on Table 3-II.

TABLE 3-II
DIMENSIONS OF PLASTIC PNEUMATIC TUBES

Length (in inches)		I.D. Diameter (in inches)	Remarks
Per Section	Total		
1. Diverging tube	(7 sections)		
16		1/2	Inlet end
6		5/8	
6		3/4	
6		7/8	
6		1	
6		1-1/4	
10		1-1/2	Collector end
	56		
2. Constant diameter tube			
	56	1/2	

A single stage vaneaxial blower made by Globe Industries and rated at 37 cfm for 1.5 inches of water pressure rise was used to provide the flow in the tube. The rotational speed of the blower as a function of input voltage was determined for the blower by itself with the adapter, blower and adapter attached to the tube, and for the complete blower, adapter,

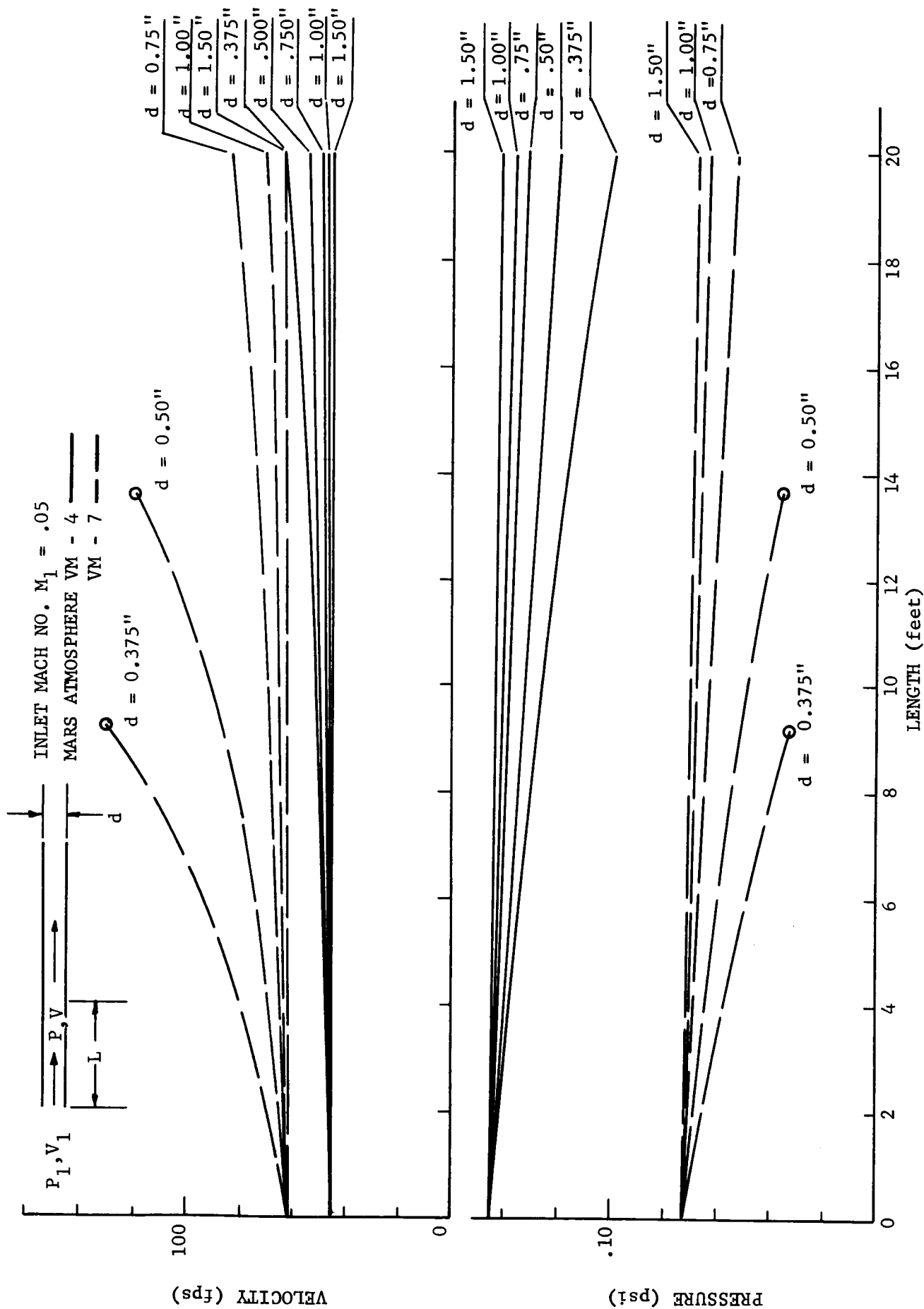


FIGURE 3-27 COMPRESSIBLE FLOW IN TUBES

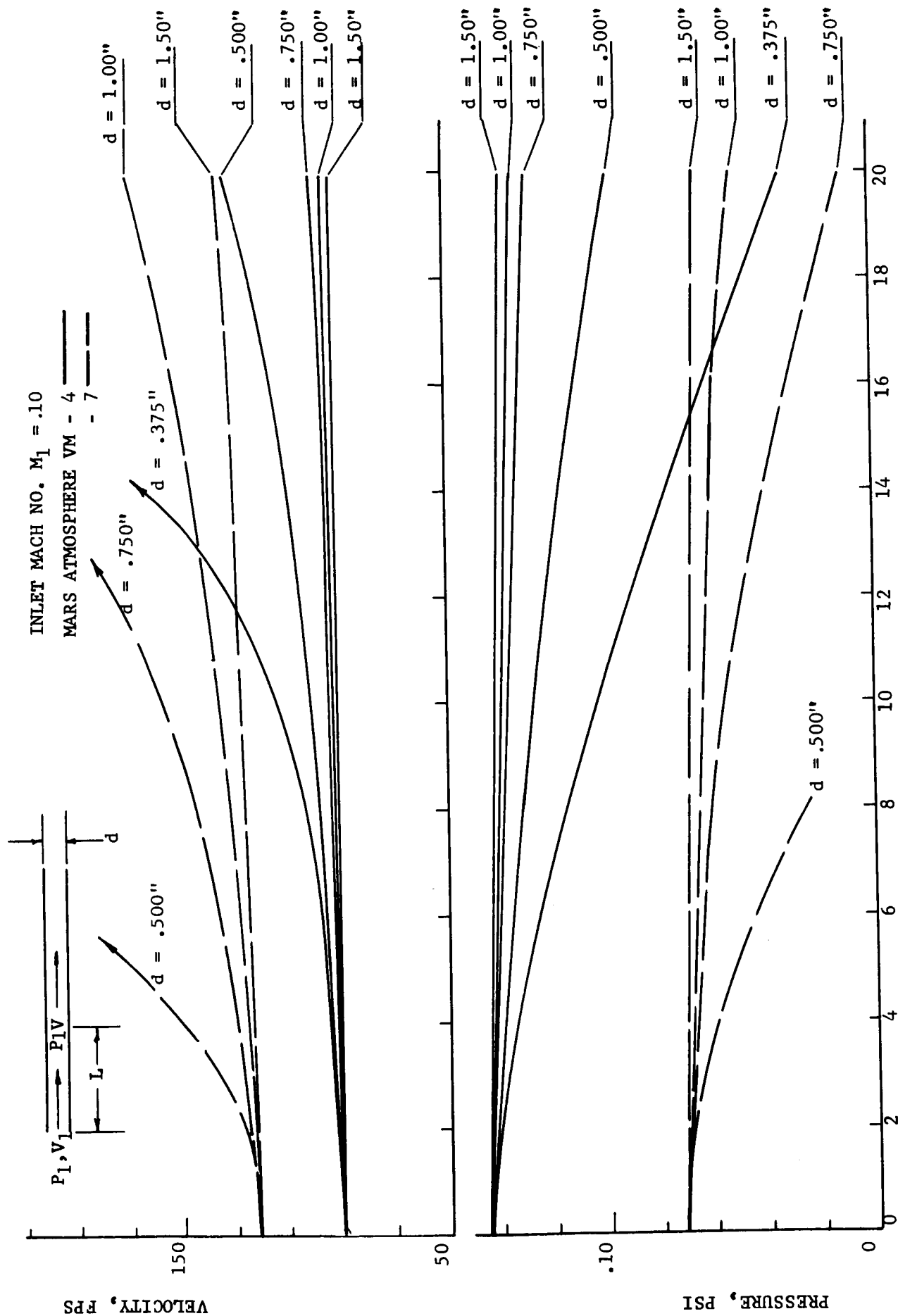


FIGURE 3-28 COMPRESSIBLE FLOW IN TUBES

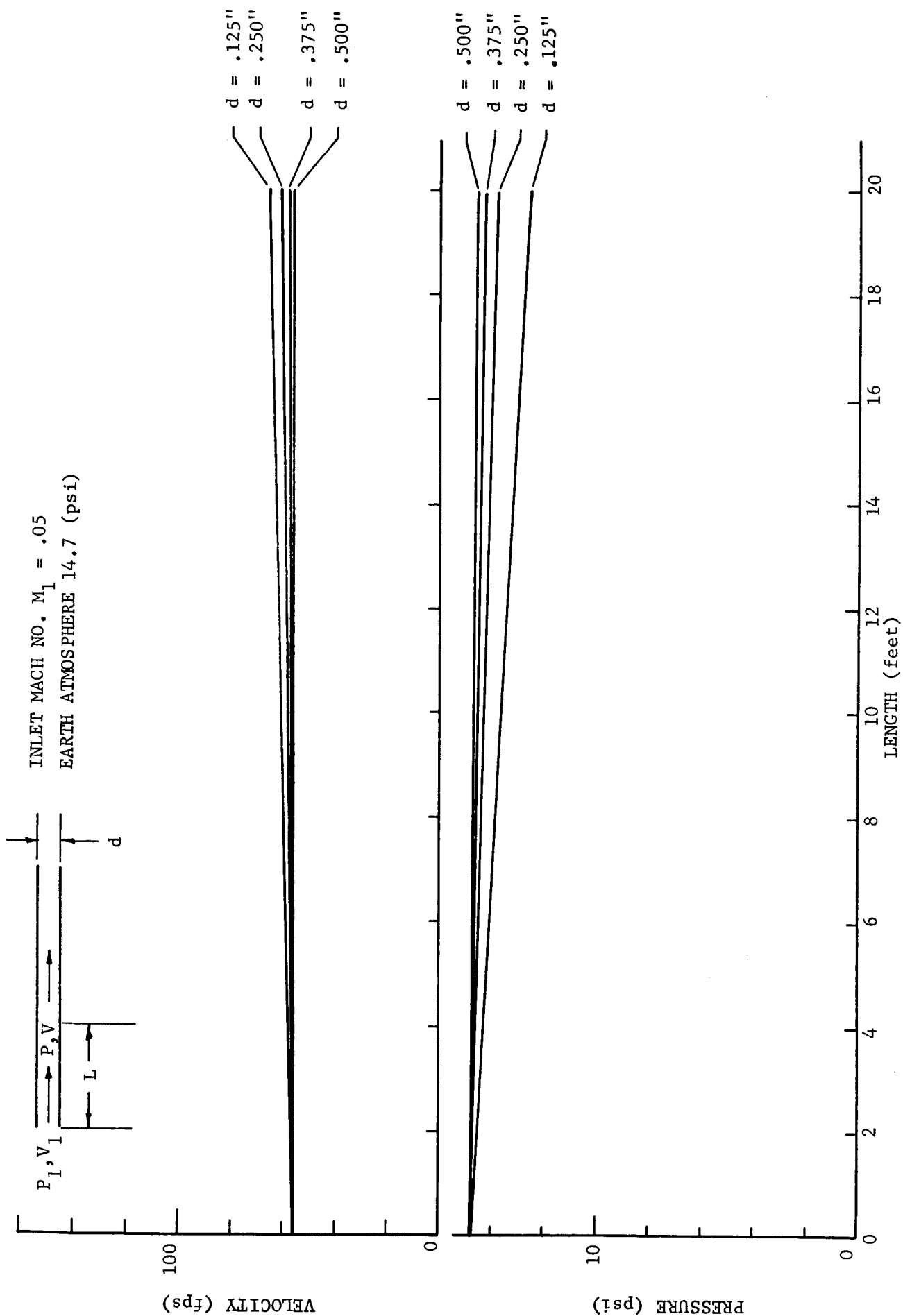


FIGURE 3-29 COMPRESSIBLE FLOW IN TUBES

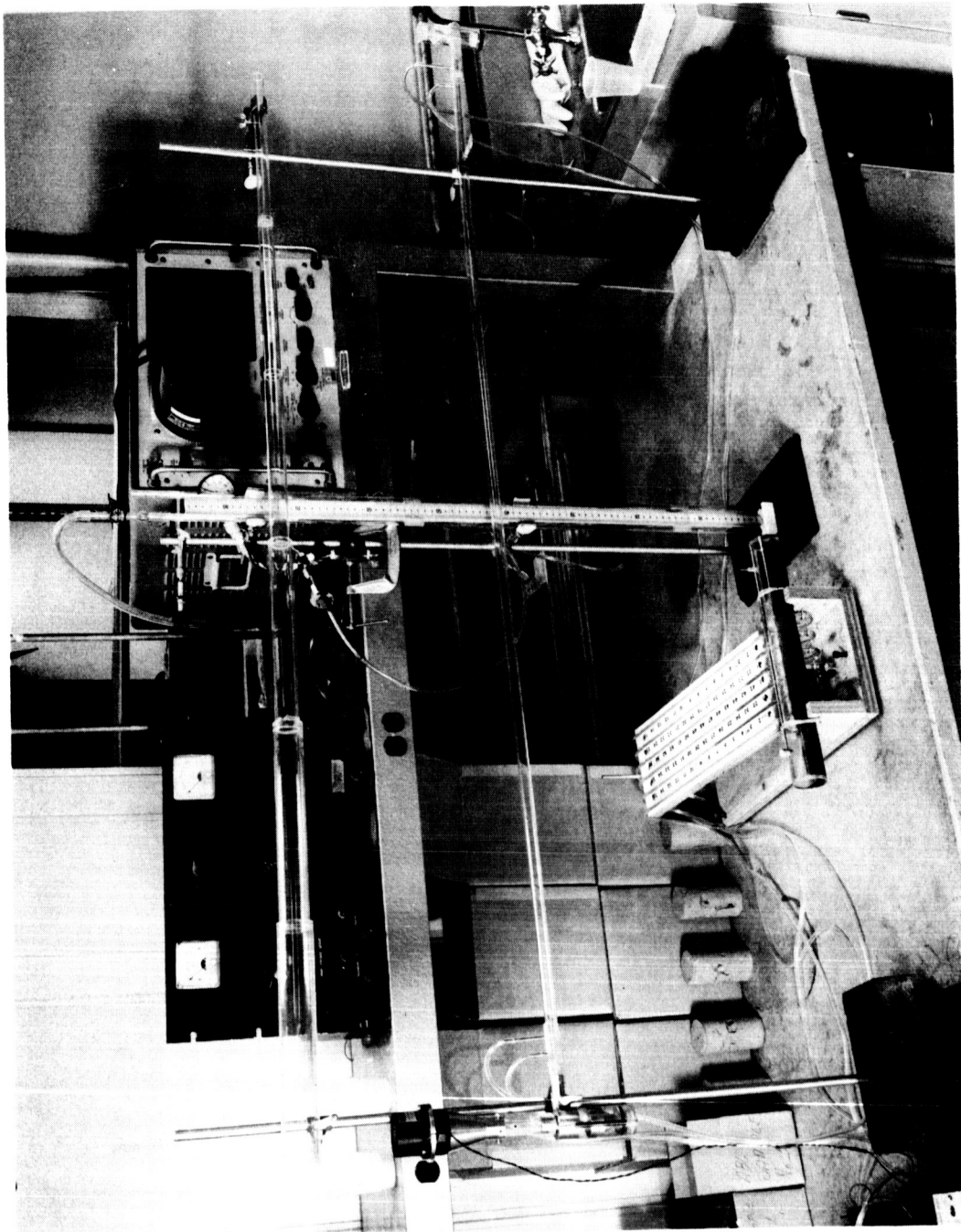


FIGURE 3-30 PNEUMATIC TUBE AND MANOMETER TEST SETUP

tube, and cyclone collector combination. These data are shown in Figure 3-31 from which it can be seen that the blower speed varies linearly with voltage and absorbs a maximum of 17 watts of power. The blower is 2 inches in diameter and is connected through an adapter to a 0.5 inch inside diameter acrylic plastic tube. Static pressure and dynamic pressure ports are located at the tube inlet and 5 feet from the inlet. A static pressure port is also located on the outlet side of the cyclone collector. These are connected to an inclined manometer. At the rated voltage of 26 volts, a velocity traverse was made at the inlet and the 5-foot station. No variation in velocity across the tube diameter could be detected at the inlet. The average flow velocity at this point was 36 fps. A traverse at the 5-foot station indicated a velocity profile shown in Figure 3-32.

These velocities produced a flow of approximately 3 cfm which is about one twelfth of the operating capacity quoted by Globe Industries for this blower. This is probably due to the fact that the cross sectional area of the tube is one ninth of the disk area of the blower causing the tube to be the limiting factor in establishing the flow rate. The inlet Mach number for this velocity is 0.033 which is low enough to consider the flow to be incompressible. At the low pressure of 5 millibars, the dynamic pressure will be reduced by the ratio of atmospheric densities resulting in an estimated pressure differential of 0.004 inches of H_2O . This is below the sensitivity of the manometer; however, at these pressures and velocities the Reynold's number is so low that the flow will not become turbulent allowing the flow velocity to be determined from the pressure drop along the tube.

To verify the feasibility of using a single stage vaneaxial blower at 5 millibars, additional calculations were made assuming incompressible flow. The pressure drop as a function of tube diameter for a 10 foot long transport tube as well as the required compression ratios are shown in Figure 3-33. It is seen that although the pressure drop is not greatly different for Earth and Mars atmospheres, the required compression ratio across the pump is considerably greater for the reduced atmosphere. The maximum compression ratio across a single stage of stator and rotor blades for an axial flow compressor is about 1.2:1. The compression ratio across the Globe blower is well below this value; however, the required transport velocity is not clearly defined depending on the soil particle density and size that it is desired to transport. Alternative pumping methods also are available which should achieve the requisite compression ratio.

When the test apparatus had been set up as displayed in Figure 3-30, a series of tests were conducted under earth ambient conditions to develop the characteristics of the equipment and operational techniques to be used in the simulation chamber. Soil transport efficiency was found to be high at velocities in excess of 20 fps for the overall system. Difficulty was experienced in dispensing the very fine soil particles in a size range

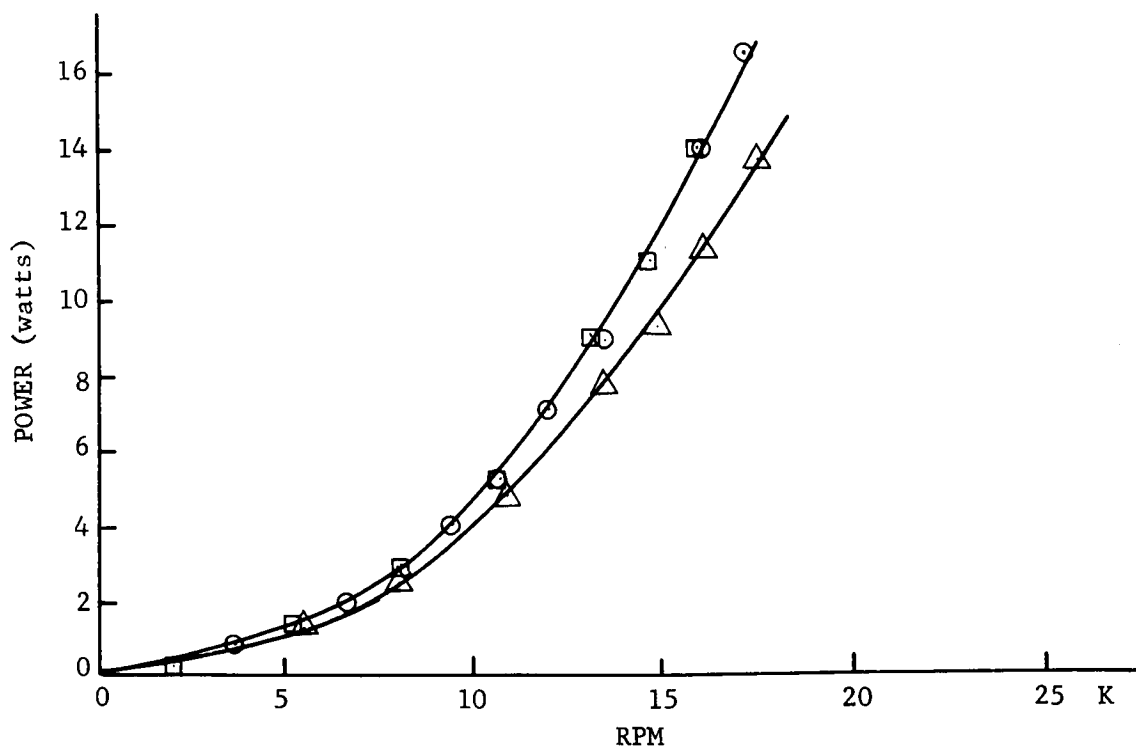
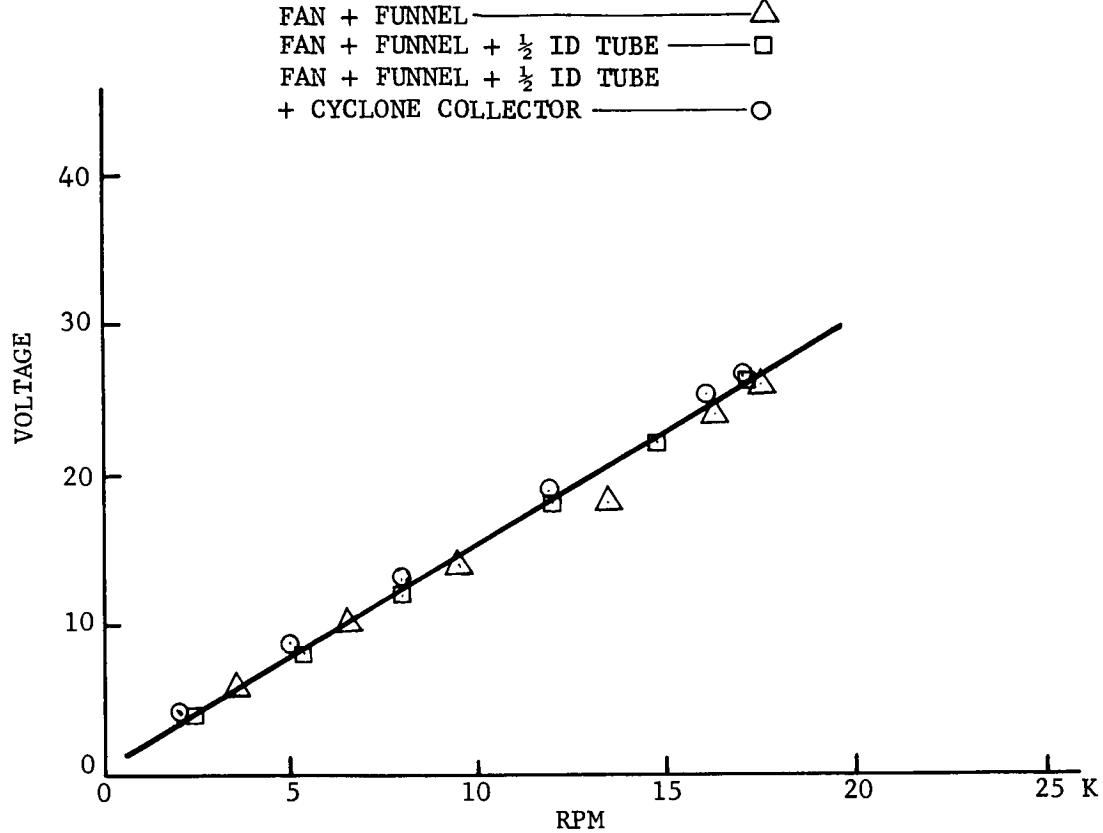


FIGURE 3-31 VANEAXIAL BLOWER, VOLTAGE AND POWER VARIATION WITH SPEED

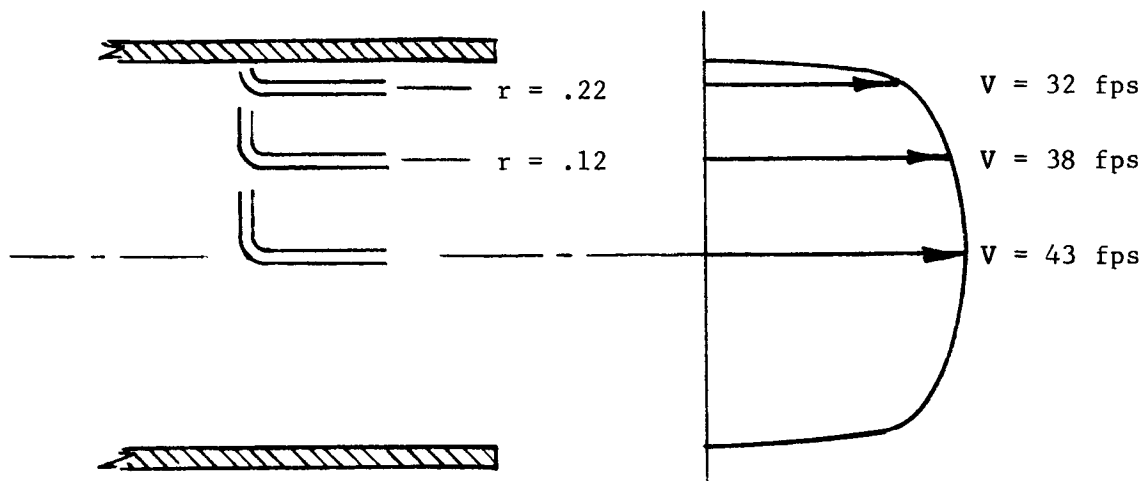


FIGURE 4.

FIGURE 3-32 VELOCITY PROFILE IN TUBE

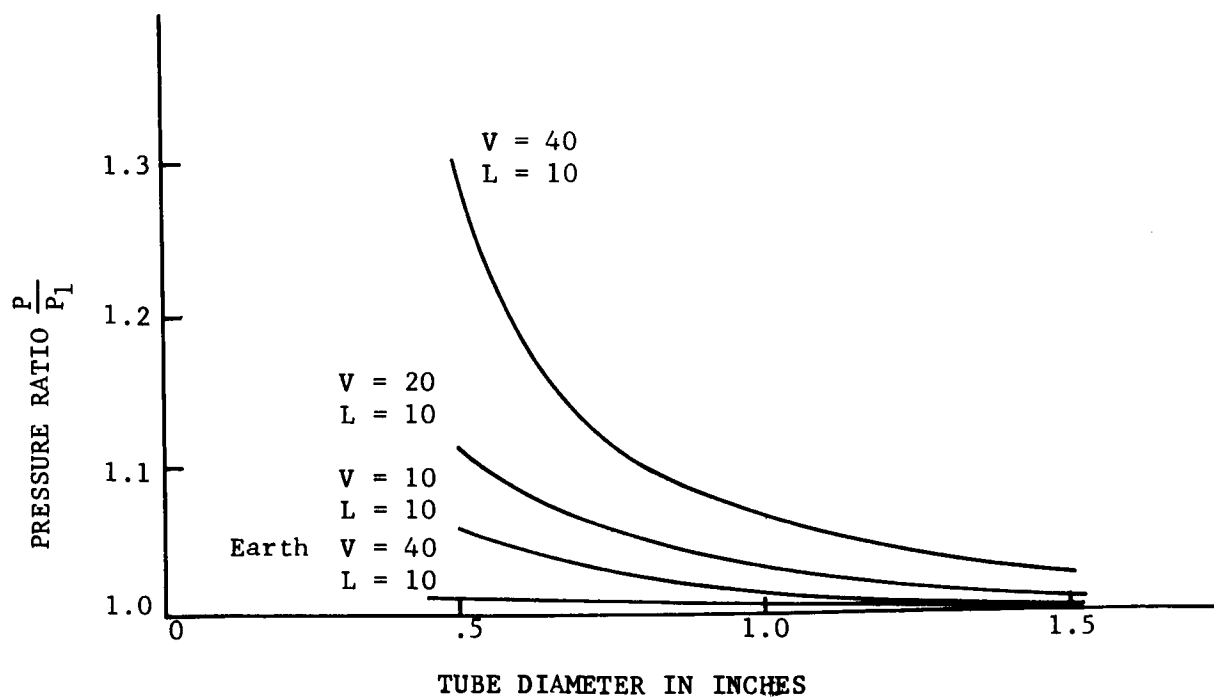
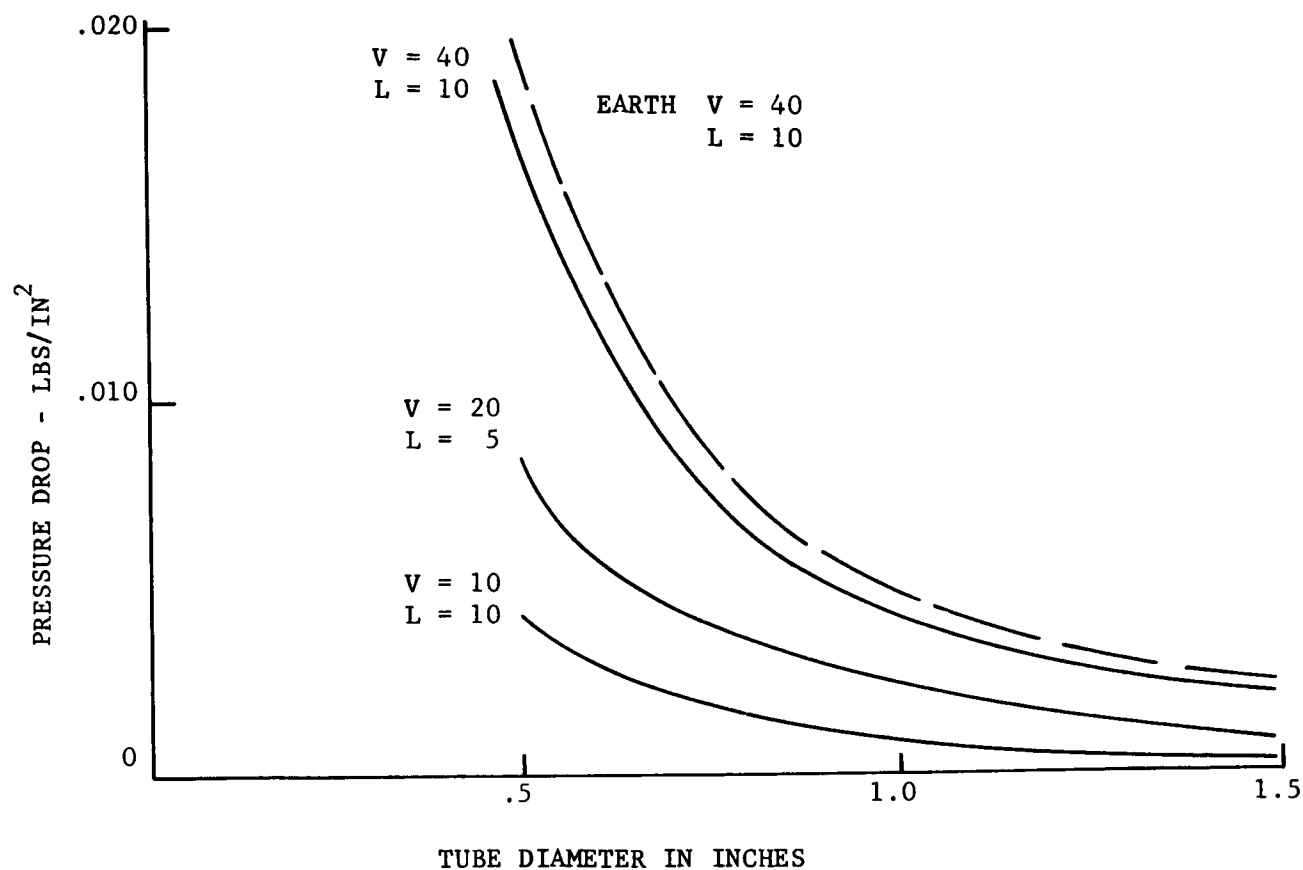


FIGURE 3-33 PRESSURE DROP AND REQUIRED PRESSURE RATIO FOR PNEUMATIC TRANSPORT

of 44 microns and less. This size of particle tends to compact and agglomerate in large masses which are too large to transport. Various methods of dispensing the soil were evaluated.

The initial dispenser consisted of a conical chamber through which the soil flowed by gravity. Frequent stoppage occurred so a vibrator was attached to the dispenser. With the vibrator, all soil particle sizes except that below 44 microns were dispensed. The very fine particles were simply compacted and vibration did not maintain a flow of soil past the inlet. Since, the vibrator produced standing waves in the transport tube, it was felt that this technique for dispensing was questionable. An aerosolizing jet dispenser proved to be unsatisfactory for the breadboard set-up since the required flow through the jet seriously affected the inlet conditions while it did not completely disperse the agglomeration of fine particles.

A mechanical dispenser consisting of a rotating wheel running with the lower half immersed in the soil proved to be the most consistent and smoothest dispenser tried. The very fine particles were still thrown occasionally as an agglomerated mass which shatters when it hits the dispenser wall. This results in a burst of soil being transported down the tube. This dispenser is shown schematically in Figure 3-34 and was used in the remaining tests. A series of runs were also made using a mix of the particle size cuts on an equal weight basis for each size range. It is interesting to note that vibration induced separation of the fine particles in the dispenser. A similar separation was noted in the cyclone collector cup where the very fine particles built-up on the walls, while the coarse material collected at the bottom beneath the conical outlet.

A few test runs were also made in the diverging tube. In this case larger particles were transported through the tube without settling out in appreciable quantities even though the velocity fell to 10 feet per second in the larger sections of the tube. Finer particle sizes were noted to settle in larger amounts with no tendency to dislodge after settling. Close observation of the transport indicates that the larger particles are being transported by saltation, i.e., bouncing along the tube. The tendency of the smaller particles to settle out and remain undisturbed can be explained by Bagnold's observation that as the surface roughness decreases, below some minimum value, the velocity required to initiate movement increases rapidly. This effect was further substantiated when the soil mixture was used. The fine particles again settled in the larger tube sections but the impact of the saltating larger sized particles could be seen to dislodge a small area of the fine particles at the point of impact, thereby aiding the movement of the fine particles.

The design and fabrication of an effective soil dispenser terminated the operational technique tests. Formal engineering tests of the breadboard

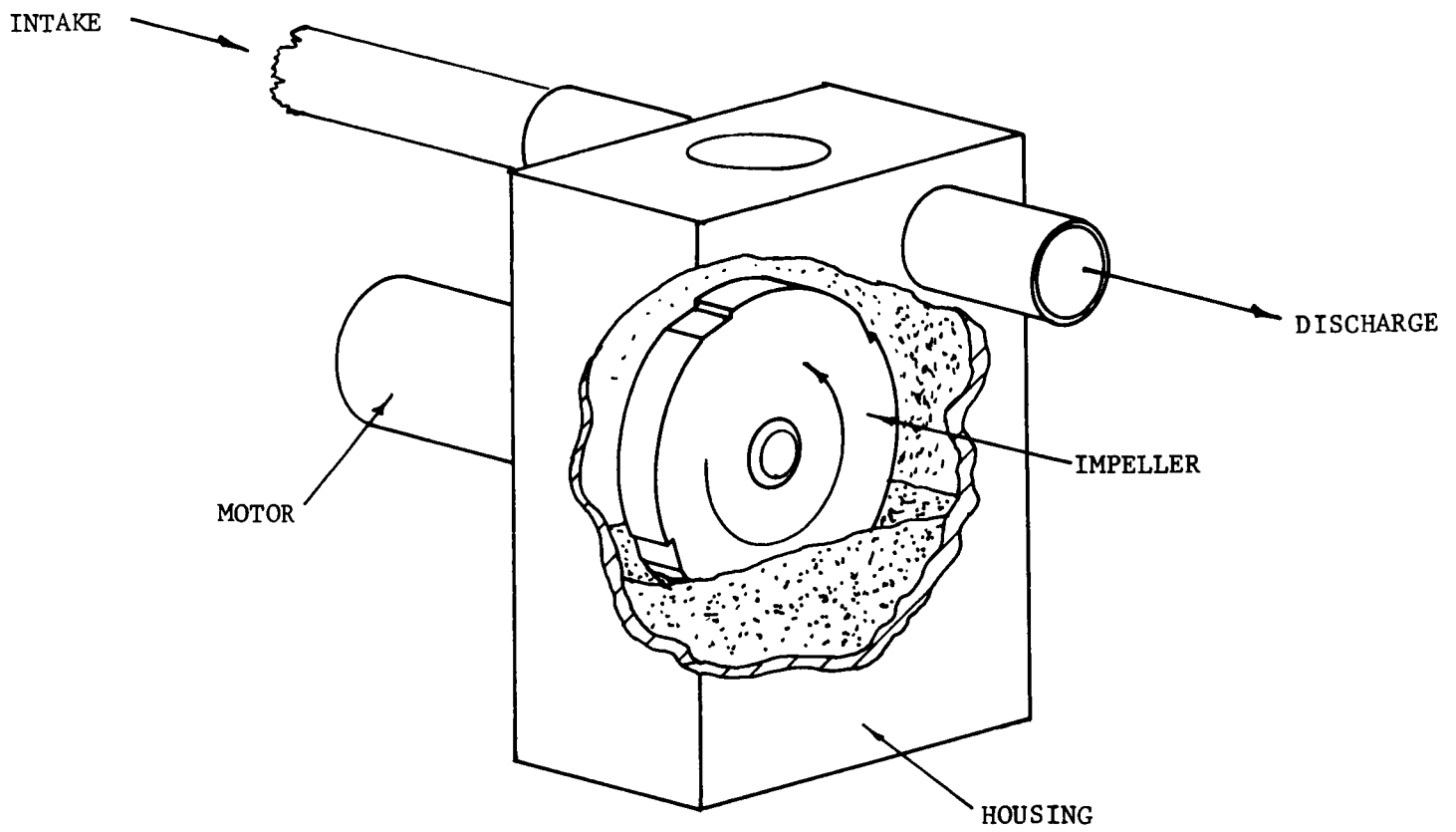


FIGURE 3-34 SOIL SAMPLE DISPENSER

transport subsystem were undertaken per the test matrix denoted in Table 3-III. The general purposes of these tests were to determine the design parameters associated with the pneumatic transport and terminal collection of soil particles within constant diameter and diverging tubes using a blower/cyclone collector system. These tests were conducted in two phases, one in ambient earth atmosphere and one in a reduced (5 millibar) atmosphere. The quantitative variables are more easily measured and compared to theory under ambient earth conditions. Also, these tests were needed to predict and evaluate the performance at reduced pressures for a system which is also expected to operate at ambient conditions.

Under ambient atmospheric conditions the following parameters were measured or observed:

- (1) Weight of soil sample collected.
- (2) Soil particle size collection efficiency.
- (3) Flow velocity required to transport.
- (4) Pressure drop through system.

Four specifically sized soil cuts were used for efficiency variation tests and a fifth material, composed of a mixture of the four cuts, was tested for total transport response. The sized soils included:

- (1) $63\mu < d < 44\mu$ derived from commercial Nevada 60 sand.
- (2) $125\mu < d < 63\mu$ derived from commercial Nevada 60 sand.
- (3) $250\mu < d < 125\mu$ derived from commercial Nevada 60 sand.
- (4) $500\mu < d < 250\mu$ derived from commercial Nevada 60 sand.
- (5) $500\mu < d < 44\mu$ being (1) + (2) + (3) + (4)

The flow velocities used were 10, 20, 30, and 40 feet per second at the inlet. On each run, a known amount (15 grams) of soil for a given particle size was placed in the dispenser. After the flow was allowed to stabilize at the appropriate velocity, soil was presented at the inlet by the dispenser. After the run was completed, the amount of soil remaining in the dispenser, the amount remaining in the tube, and the amount collected by the cyclone collector were retrieved and weighed.

TABLE 3-II I

PNEUMATIC TRANSPORT TEST MATRIX

TEST CONFIGURATION	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	36	37	38	
CONFIGURATION PARAMETERS																																						
BLOWER CONFIGURATION:																																						
SINGLE BLOWER, 37 cfm (1ATMOS)																																						
DUAL PUSH-PULL BLOWERS																																						
PNEUMATIC TUBE CONFIGURATION:																																						
CONSTANT DIAMETER																																						
DIVERGING																																						
CYCLONE COLLECTOR CONFIGURATION:																																						
(1) STANDARD (1.75 INCH SPACER)																																						
(2) DEFLECTOR ADDED																																						
(3) SPACER REDUCED (1.25 INCH SPACER)																																						
AMBIENT TEST PRESSURE:																																						
1013 mb																																						
26																																						
16.3																																						
10.8																																						
4.5 mb																																						
SOIL SAMPLE GRAIN SIZE:																																						
44µ<4																																						
44µ<4<63µ																																						
63µ<4<125µ																																						
125µ<4<250µ																																						
250µ<4<500µ																																						
44µ<4<500µ																																						
GROUND NYLON (63µ<4<500µ)																																						
DYNAMIC PARAMETERS																																						
AVERAGE FLOW VELOCITY:																																						
40 fps																																						
30																																						
20																																						
15																																						
10 fps																					</																	

NOTE: (1) Plastic spacer inserted between top of collecting funnel and vortex generator measures 1.75 inches.
 (2) Plastic disk sealed by rubber O-ring to wall of collecting cut inserted at narrow end of funnel.
 (3) Plastic spacer reduced to 1.25 inches.
 These devices, inserted to improve collector efficiency, did not appear to aid sample collection.

The system efficiency was based on the amount of soil collected by the cyclone collector of the total available in the dispenser. The cyclone collector efficiency was determined based on the amount of soil in the collector compared to that which might have been collected. The soil remaining in the tube and dispenser were not charged to the cyclone collector. The efficiency was calculated as follows:

$$e_c = \frac{W_{\text{collected}}}{W_{\text{total}} - W_{\text{tube}} - W_{\text{dispenser}}}$$

Therefore, only that portion of soil which passed through the collector was charged against the cyclone collector's efficiency.

The purposes for testing the pneumatic transport tubes in levels simulating the range of atmospheric pressures assumed for Mars were:

- (1) To determine if the theoretical predictions of the ability of pneumatic tubes to transport soil particles in the 63μ to 125μ range can be borne out at pressure levels of 5, 10 and 15 mb and to quantitatively test their efficiency at these pressures.
- (2) To assess the relative merits of the diverging and constant diameter pneumatic tubes in reduced Mars-like atmospheres.
- (3) To investigate the relative efficiency of a single blower versus a dual stage-push-pull system.
- (4) To illustrate the sample collecting efficiency of the best pneumatic transport configuration for acquiring silicate samples under simulated Martian gravitational conditions.

And the parameters that were measured or observed during reduced pressure tests were the following:

- (1) Visually determine distance that sand grains travel down the pneumatic tube.
- (2) Weight of soil sample remaining in the pneumatic tube.
- (3) Weight of soil sample collected.

- (4) Estimate of particle size distribution in soil sample collected.
- (5) Voltage and current input (to detect power variations in blower operation).

The tests were conducted in four phases utilizing specifically sized Nevada sand as the soil medium:

- (1) Diverging diameter tube with cyclone collector blower only.
- (2) Diverging diameter tube with push-pull blower system.
- (3) Constant diameter tube with cyclone collector blower only.
- (4) Constant diameter tube with push-pull blower system.

When the relative merits of these four phases were determined, the best configuration was tested using ground nylon in order to simulate the density of silicates on Mars.

A rotating disk soil dispenser was mounted directly on the inlet end of the tube when tests were conducted that utilized the single stage cyclone collector blower. When testing the push-pull blower system set up, a 1-1/8" diameter Globe Industries blower was sandwiched between two conical adapters, one leading to the soil dispenser and the other attached to the inlet end of the pneumatic tube. The material used on the four test phases consisted of Nevada No. 60 sand, hand sieved to obtain the range, $63\mu < d < 125\mu$. The simulated Martian silicate material consisted of ground nylon (density 1.1 g/cm^3) having the following grain size distribution:

100% passed through No. 35 standard sieve

52% retained on No. 60 (250μ)

37% retained on No. 120 (125μ)

10% retained on No. 230 (63μ)

1% retained on No. 325 (44μ)

Grain size range: $100\% < 500\mu$ and $99\% > 63\mu$

Mean grain size: $\approx 250\mu$ (medium grained nylon sand)

The Nevada sand is wholly composed of rounded to subrounded, equidimensional grains. The ground nylon, on the other hand, exhibits a variety of grain shapes as follows:

Equidimensional grains	- 75%	Subrounded	55%
		Subangular to wedge-shaped	20%
Fibers to rod-like grains	- 20%	Fibers to unraveled fibrous bundles	15%
		Rod-like	5%
Platy grains	- 5%	Twisted "fruit peel-shaped"	4%
		Flat plates	1%

The above description applies to the gross sample; as the grain size approaches 63μ , the percentage of angular equidimensional grains approaches 100%.

The NASA Mars simulation chamber was utilized to perform the tests. It is sufficiently large to contain the 56-inch long pneumatic tubes and appended soil dispensing and collecting gear.

The results of the tests of breadboard models of constant diameter and diverging pneumatic transport tubes can be initially summarized and then discussed in detail in succeeding paragraphs. Tests of the constant diameter tube under room ambient pressure showed that the overall system efficiency was better than 90 percent for all flow velocities above 20 fps and for critical grain sizes (63μ - 250μ) approached 90 percent for flow velocities above 10 fps. These results are summarized on Figure 3-35. At the same time the collector efficiency was essentially 100 percent for all flow velocities above 10 fps as denoted in Figure 3-36. Tests in the diverging tube at flow velocities of 40 fps demonstrated that the collector efficiency was essentially 100 percent but the overall system efficiency was less than 10 percent, therefore, further tests at lower flow velocities were abandoned.

The tests in the constant 0.5 inch diameter acrylic plastic tube at room conditions yielded the following conclusions:

- (1) The overall system efficiency is 70 percent or greater for flow velocities of 20 feet per second for all particle size ranges tested (44μ - 500μ).
- (2) The cyclone collector efficiency approached 100 percent for flow velocities above 20 feet per second.

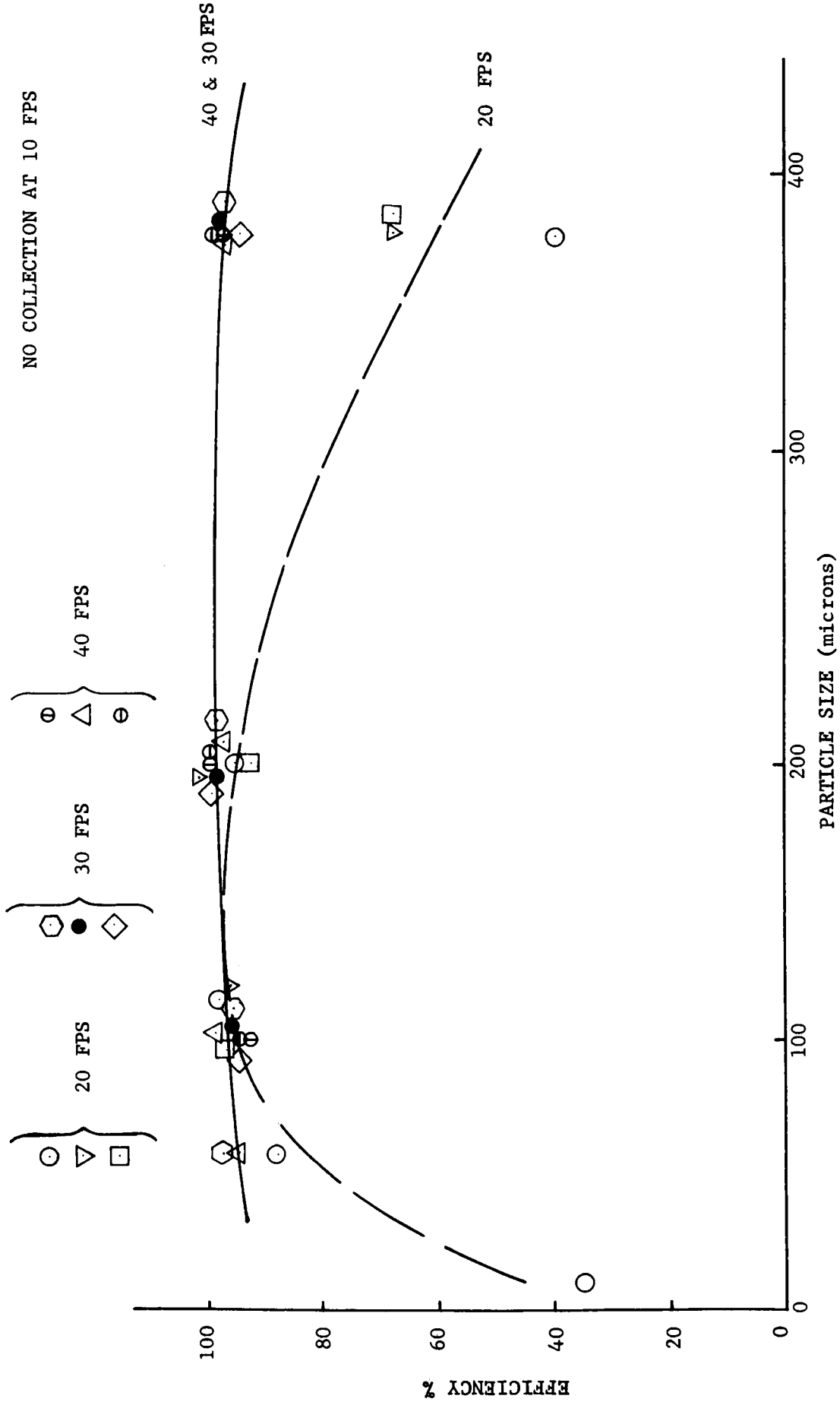


FIGURE 3-35 PNEUMATIC SOIL TRANSPORT ABL OVERALL SYSTEM EFFICIENCY

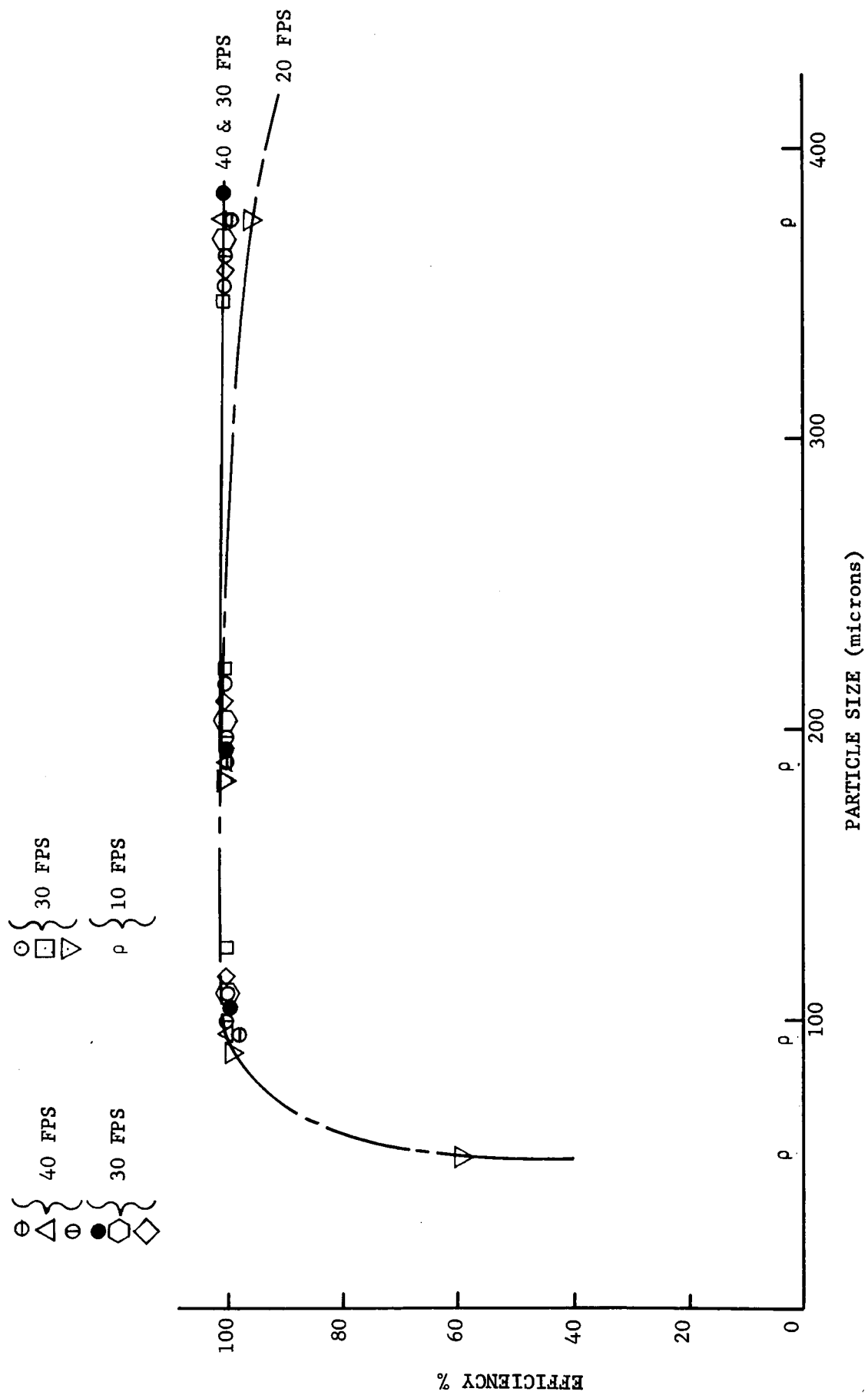


FIGURE 3-36 PNEUMATIC SOIL TRANSPORT CYCLONE COLLECTOR EFFICIENCY

- (3) Virtually no soil was transported at 10 feet per second. Under these conditions it is difficult to assess the cyclone collector's efficiency since no measurable amount was transported.

And tests on the diverging diameter tube fabricated to correspond to the internal dimensions of an extended telescoping boom and operating at sea-level pressures demonstrated that:

- (1) The cyclone collector efficiency continued to be good.
- (2) The overall system efficiency, on the other hand, was less than 10 percent for maximum flow velocities capable of attainment within the system.

Velocity traverses over the cross-sectional area of the 0.5 inch pneumatic tube were made at flow velocities of 40, 30, 20 and 10 fps at a pressure level of one atmosphere. The results are summarized on Figure 3-37. These traverses were eliminated for the diverging tube due to its low collecting efficiency.

Both constant diameter and diverging breadboarded pneumatic tubes were tested within the Mars simulation chamber at reduced pressure levels of 26, 16.3, 10.8 and 4.5 millibars. Table 3-IV summarizes the results of the reduced pressure tests. The series of tests were conducted with a specifically sized fraction of Nevada No. 60 sand, 63μ to 125μ , in order to determine relative performances of pneumatic tubes at reduced pressure levels. The tests were not so statistically numerous as to warrant graphical illustration, but they did augment the following conceptual design goals:

- (1) A pneumatic transport tube will transport measurable quantities of quartz sand at 10 mb atmospheric pressure and gram-size amounts at 15 mb.
- (2) The constant diameter pneumatic transport tube equipped with a two-stage, push-pull blower system is the most effective configuration of those tested at atmospheric pressure levels of 5 to 15 mb.

A trace of sand was actually transported down the pneumatic tubes for distances up to 3.75 ft at 4.5 ± 0.5 mb pressure (see tests Nos. 26 and 29). This fact leads to two conclusions:

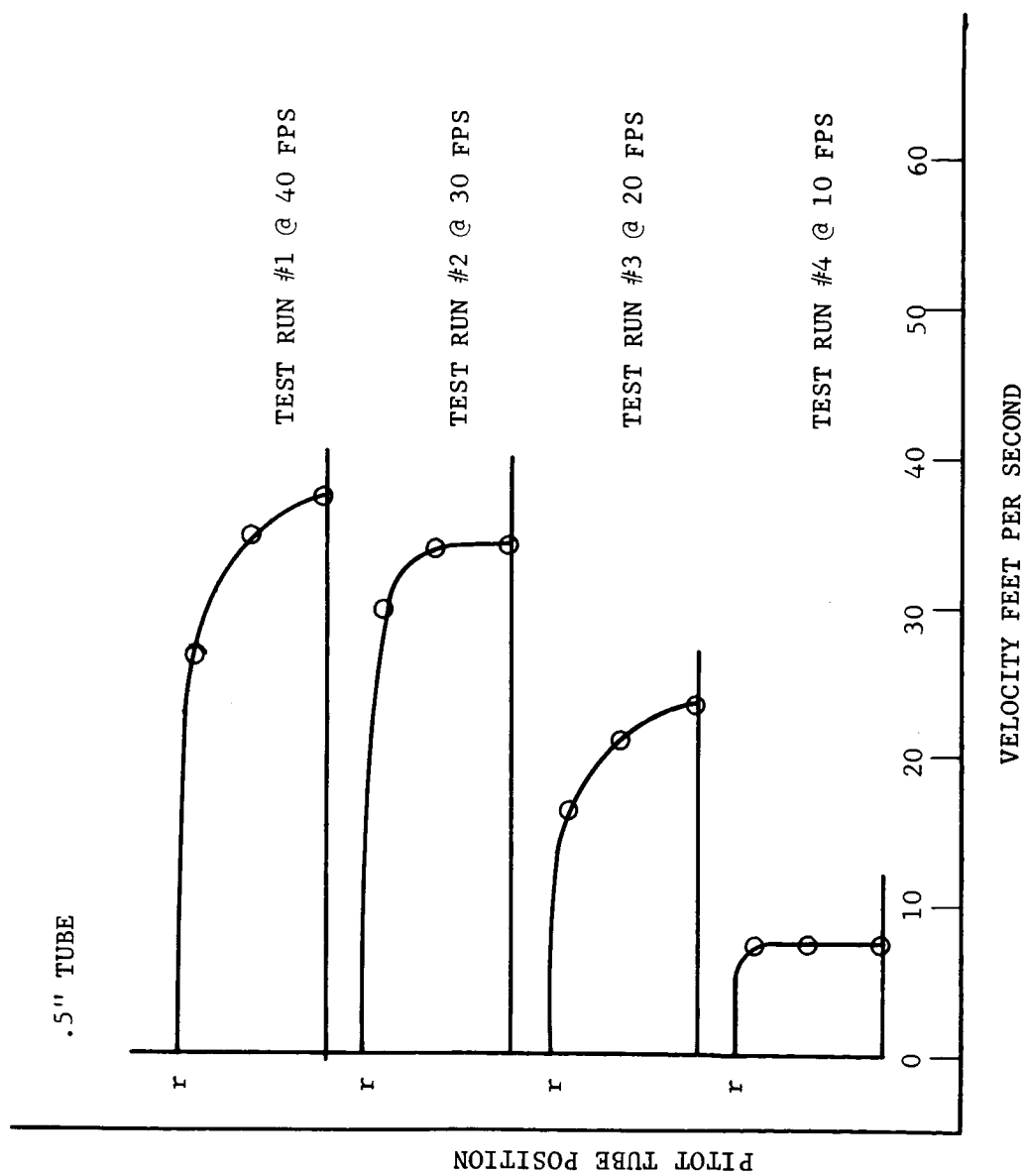


FIGURE 3-37 PNEUMATIC TRANSPORT VELOCITY PROFILE

REDUCED PRESSURE TESTS OF PNEUMATIC TUBE TRANSPORT SYSTEMS

3-62

- (1) The theory used to predict the pneumatic transport of soil particles by blowers at low ambient atmospheric pressures was essentially correct.
- (2) Multistage blowers will undoubtedly transport adequate samples at pressures as low as 5 mb.

There was no grain size differentiation evident in the sample collected, regardless of tube configuration, blower system utilized, vacuum level, or quantity of material collected. In all cases, both 63 μ and 125 μ materials were either collected or were present in the pneumatic tubes.

Ground nylon does not appear to be a practical model for the gravimetric simulation of Martian silicates, for the following reasons:

- (1) Grain shape, fibrous grains.
- (2) Surface skin greasiness.
- (3) Electrostatic sticking.

All three properties cause the material to stick to the blower rotors and rotor housing walls as well as to the walls of the acrylic transport tubes. The ground nylon tests must be viewed with this problem in mind. The sample collected at 10.8 mb (test No. 37) consisted wholly of equidimensional fines < 230 mesh (< 63 μ); that collected at 16.3 mb contained the entire range of particle sizes but, during the course of making the test, the intake blower became choked with ground nylon. A more inert soil model material must be developed and utilized for gravity simulation tests.

SECTION 4

ENGINEERING PROTOTYPE SAMPLER DEVELOPMENT

Under the terms of this contract, a portion of the task was to design, fabricate, and evaluate at least one each of an engineering prototype sampler designated as type A and type B. The type A sampler was to be capable of being deployed horizontally over the surface adjacent to the landed capsule or payload on which the sampler is mounted. The type B sampler was to be simply deployed in a vertical direction until it reached the surface immediately below the point of mounting. The two types of sampler prototypes were selected from evaluations of the conceptual design phase discussed in Section 2.0. The type A concept selected was the rotating wire brush listed as 7A in Appendix B. The type B concept selected was the conical abrasive sieve listed as 8B2 in Appendix B. The detail design, fabrication, and evaluation are discussed in the following paragraphs of this section. It should be pointed out that the primary emphasis was placed on developing engineering prototype hardware which would meet the sample requirements listed in Table 1.1 with secondary emphasis being placed on the design goals. The intent was to obtain workable hardware with a minimum of development since the fabrication schedule was fairly short. Also, the design goals are more logically satisfied during a subsequent design phase in which flight prototype hardware is fabricated after the engineering principles and mechanisms have been proven. The type B sampler is discussed first since the design and fabrication generally preceded the type A sampler.

4.1 VERTICALLY DEPLOYED CONICAL ABRASIVE SIEVE

4.1.1 DESIGN

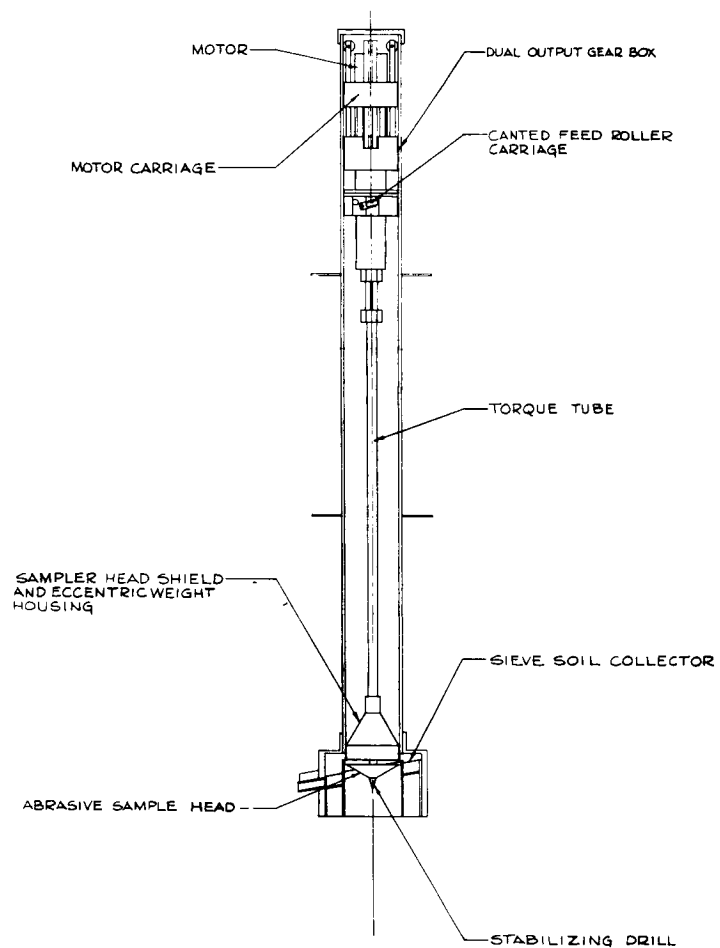
The basic design concept for this sampler as developed in the preliminary design layout (reference drawing number PD25166) utilizes a conical sampling head mounted on the end of a semi-flexible shaft which is driven by a

traveling feed mechanism inside a tube. The design objective was to make the feed mechanism a self limiting device which would automatically adjust the feed rate to the strength of the soil being sampled and the maximum applied thrust to a value not greater than 20 pounds. The sample collected in the head was to be delivered into the landed capsule by reversing the feed to retract the sampler to its stowed position at which point it would go into a high speed spin thereby transferring the sample by centrifugal force from the conical sampling head to the soil receiver. In order to achieve the high speed spin, a dual output gear train is used and engaged selectively by either an overrunning roller clutch or a friction clutch to produce the desired rotational speed.

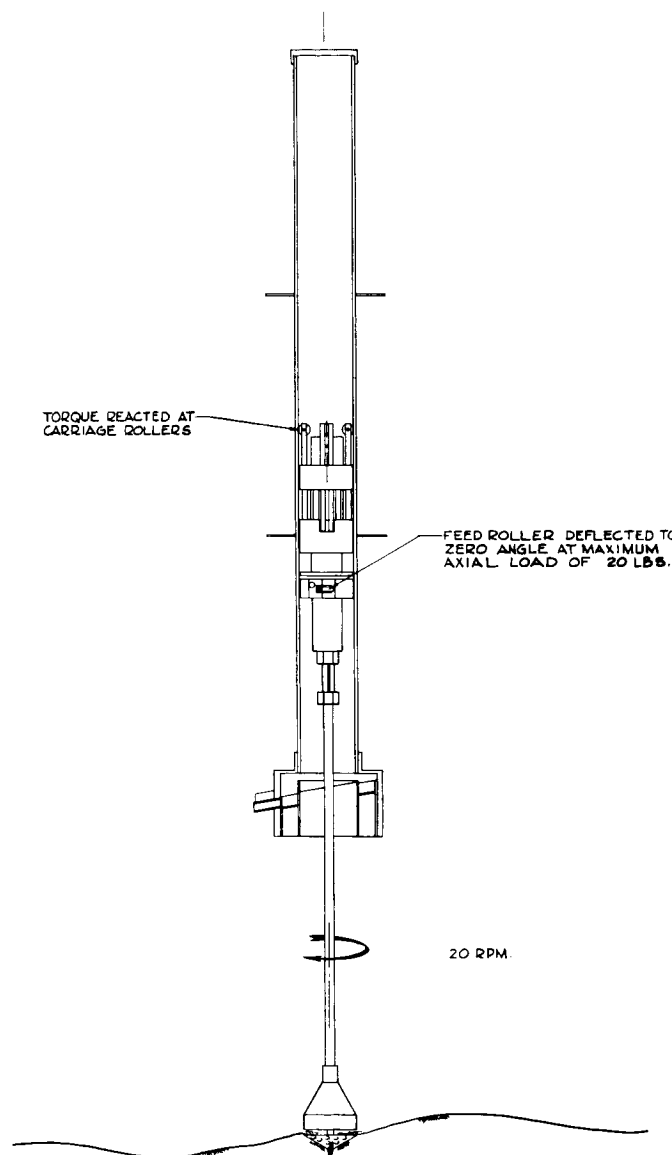
The sequence of operation for this sampler is shown schematically in figure 4-1. The entire sampler is contained inside a tube which is mounted on the landed capsule or payload. The length of the tube is dependent primarily on the distance the opening is located from the ground and the depth to which it is desired to sample. The length of the tube in this prototype was sized to fit into a 5 foot diameter capsule and to reach the surface when this capsule is situated on a surface with ± 15 degrees of slope as shown in figure 4-1. This geometry was assumed to provide some basis for selecting the tube length of 30 inches which can be any arbitrary length desired. Returning to figure 4-1, the sampler in the stowed position consists of the following major subassemblies. At the top of the tube is the motor carriage consisting of the motor and 8 spring mounted rollers. These rollers are preloaded to a normal force of 50 pounds each against the tube walls so that axial movement is allowed but rotation of the carriage is restrained. The motor torque is reacted at these rollers while at the same time allowing the carriage to travel along the tube. The gear box of the motor carriage is connected to the canted feed roller carriage through a spring loaded coupling, which keeps the friction clutch for high speed spin normally disengaged. The gear box provides a coaxial output at 20 rpm and 475 rpm. The low speed output is permanently engaged with the canted feed roller carriage which in turn is engaged to the shaft through an overrunning roller clutch. A friction clutch mounted between the end of the sampler shaft and the highspeed output of the gearbox is only engaged when the canted feed rollers are driven in reverse and the motor carriage is stopped against the upper end of the tube. The advance of the feed mechanism collapses the spring loaded coupling allowing the faces of the friction clutch to engage. The overrunning clutch allows the shaft to rotate at the higher speed. These clutches are shown schematically in the spin/dump mode of figure 4-1.

The canted feed roller mechanism operates on the principal that the canted roller describes a helical path inside the tube thus causing it to advance along the tube in a direction dependent on the direction of rotation of the carriage. The canted rollers are mounted eccentrically on the end of torsion bar supports so that as load is applied to the sampler, the torsion

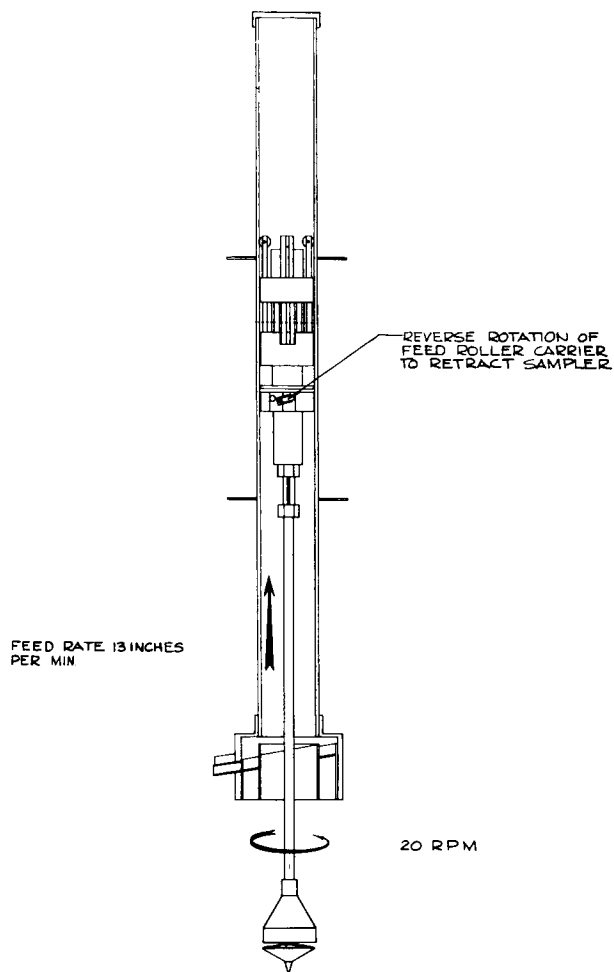
STOWED POSITION



SAMPLING MODE



RETRACTING MODE



SPIN / DUMP

MODE

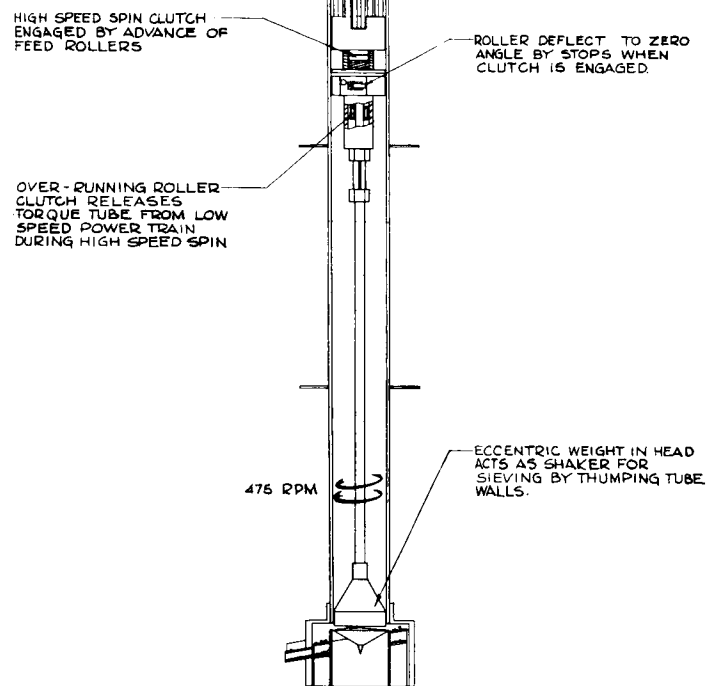


FIGURE 4-1 SCHEMATIC DIAGRAM - VERTICAL SAMPLER DEPLOYMENT

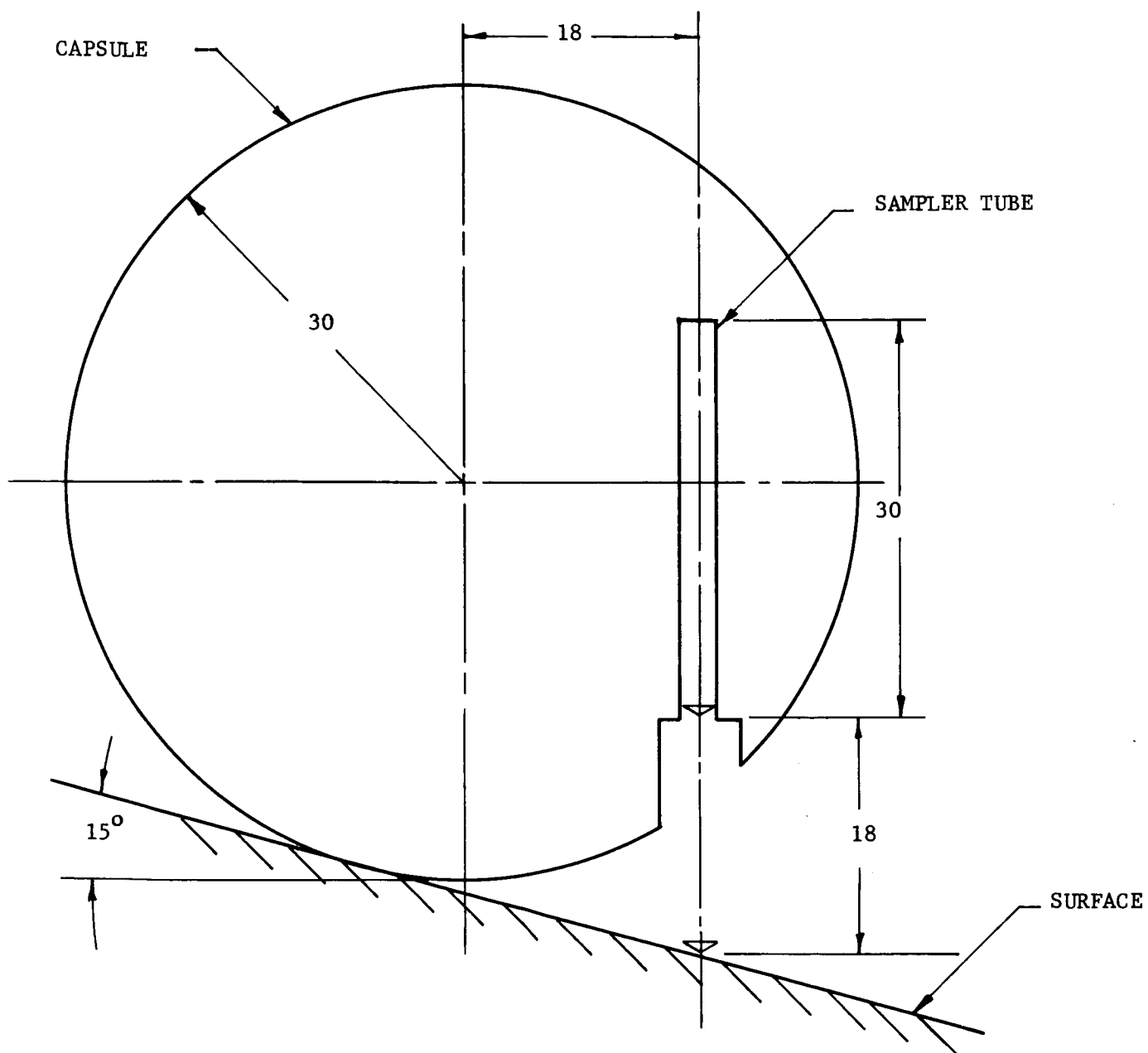


FIGURE 4-2 ASSUMED CAPSULE AND SAMPLER GEOMETRY

bar allows the rollers to deflect to a smaller cant angle thereby reducing the feed rate while simultaneously providing an axial thrust on the sampling head in proportion to the resistance offered by the soil surface encountered. The rollers are preloaded against the tube walls by placing the torsion bar in bending to provide the necessary normal force so that axial slippage is prevented when full thrust is achieved. Thus, the torsion bar supports for the canted rollers are working under a combined loading of bending, torsion, and axial compression or column action.

The sampling head is attached to a torque tube drive or shaft approximately 18 inches long. This was designed to be long and slender so that it would have a certain degree of flexibility which is desired to allow the sampling head to deflect as required to find the most favorable entry into a rubble soil model. It also allows the unbalanced sampling head to deflect far enough to strike the walls of the transport tube during the high speed spin thereby introducing agitation or vibration to assist in unloading the soil sample from the sampling head. The sampling head consists of a 2 inch diameter cone with a cone half angle of 60 degrees. At the apex of the sampling cone is located a spring loaded conical tip auger which serves to act as a center for the sampler to stabilize it during the initial contact with the surface. It is spring loaded so that the majority of the tip can retract into the conical sieve in the event a very hard surface is encountered. This will allow the conical sieve to come into contact with the surface and start collecting a sample more quickly. The complete collection head is shown in figure 4-3. Above the conical abrading sieve is located a conical shaped cover which contains a chamber in which a weight is housed to unbalance the head to induce agitation during the high speed spin dump of the soil sample, as was mentioned previously. This conical cover also serves to prevent rocks or fragments from lodging on the upper surface of the conical sieve, thereby preventing the sampling head from jamming in the tube on the sample return. This represents a variation which was not shown in the original concept as shown for sampling concept 8B-2 in Appendix B but was determined to be necessary to prevent failures due to jamming caused by pebbles or stones.

The remaining subassembly is the soil sample collector or receiver. Since the exit slit can allow small pebbles approximately one eighth of an inch in diameter to enter the sampling head from the side, it was felt necessary to incorporate a 1 millimeter mesh screen to separate the pebbles from the particle less than 1 millimeter in diameter as was stipulated in the sample requirements. The collector configuration is shown in figure 4-4. The screen and collection plate are inclined at a 15 degree angle. Thus, as the soil is delivered into the collector by centrifugal force during the spin-dump, it falls onto the screen. The agitation caused by the unbalanced sampler head also causes the soil to fall through the screen and migrate toward the delivery chutes, as well as assisting in the dump operation.

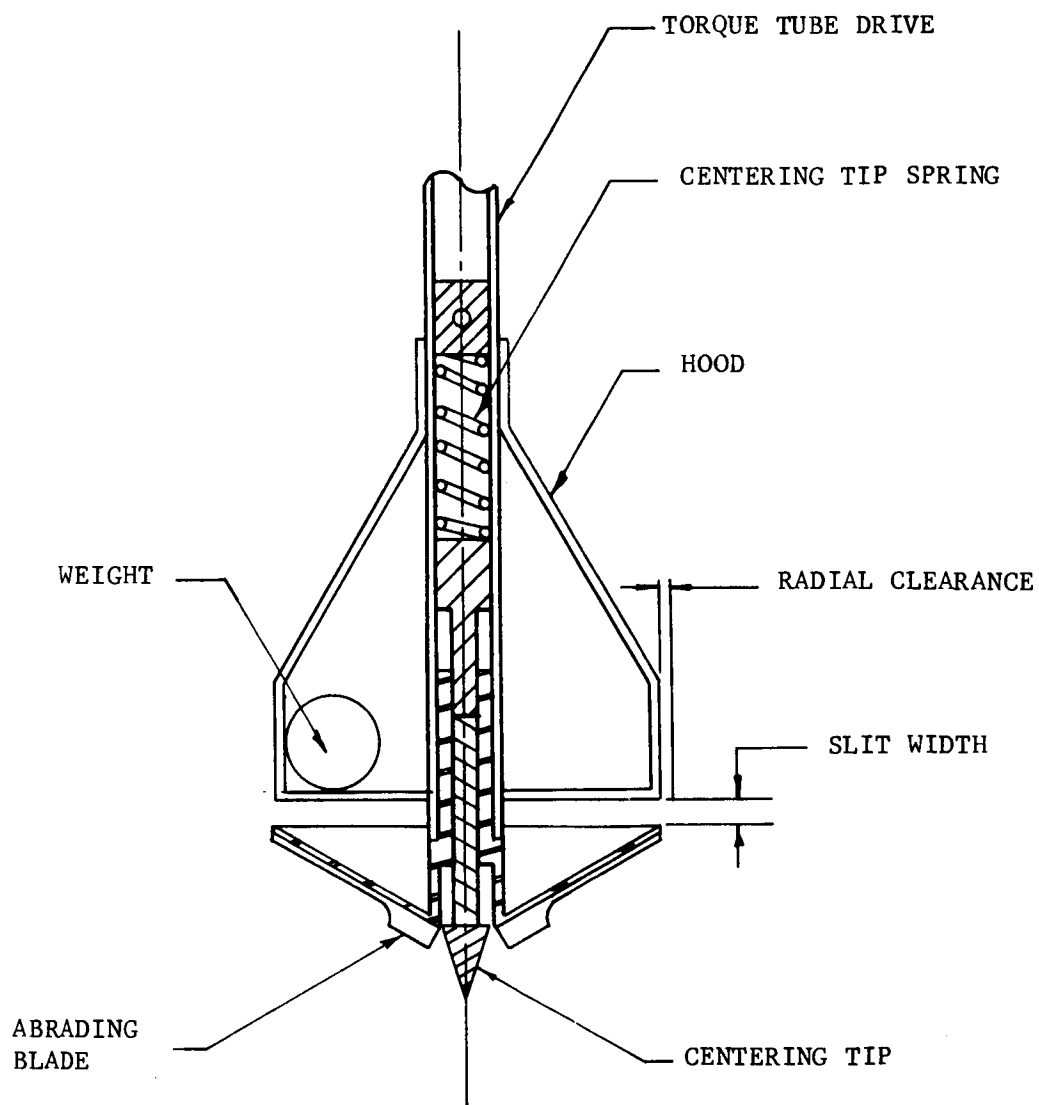


FIGURE 4-3 CONICAL ABRADING SIEVE SAMPLER HEAD CONFIGURATION

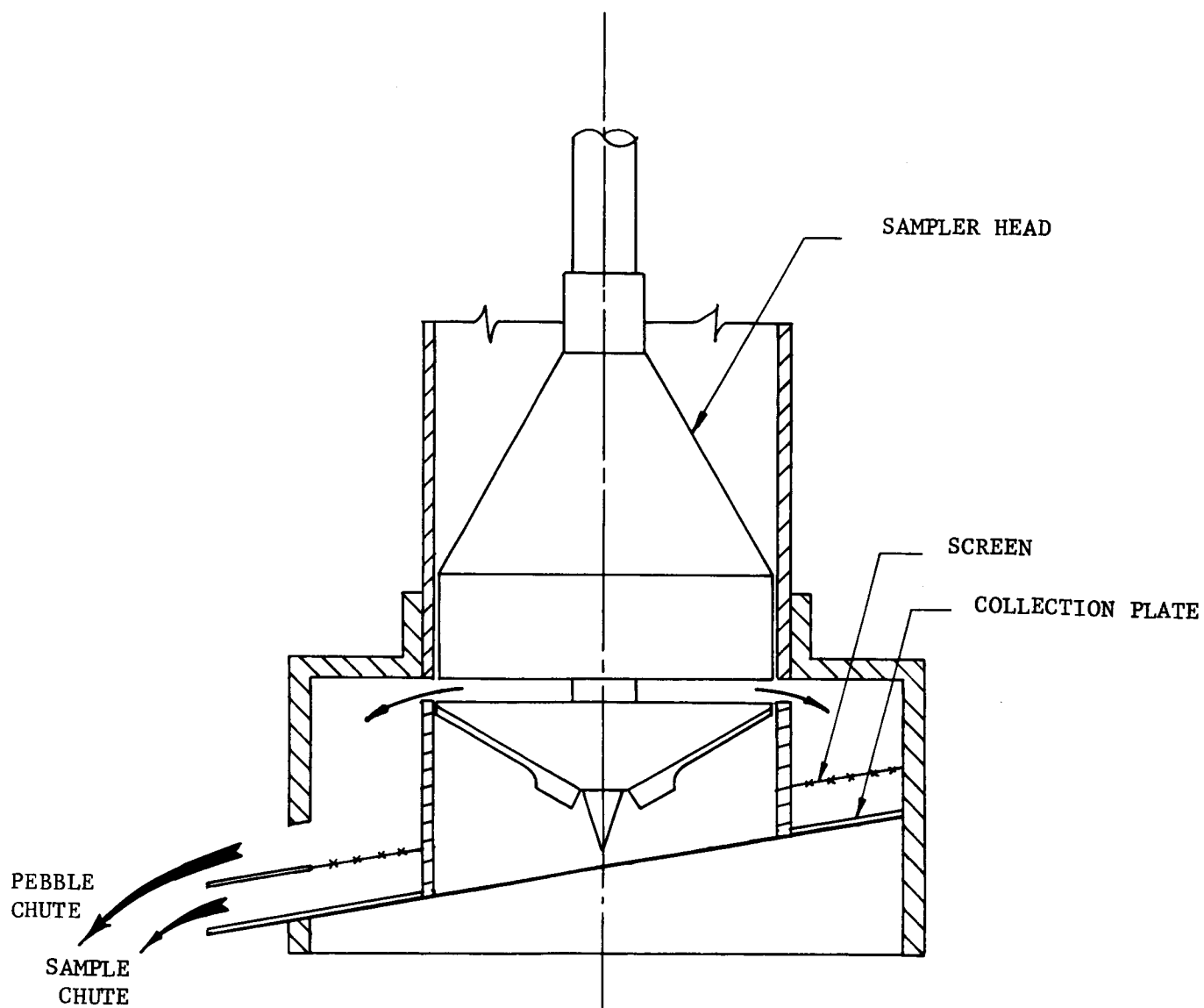


FIGURE 4-4 SAMPLE COLLECTOR CONFIGURATION

Electrical power is fed to the traveling motor through two beryllium/copper tapes wound on small drums located above the end plate of the support tube. As the sampler progresses down the tube, the tapes are deployed off the take-up drums. These drums are attached to a small spring motor which is wound up as the tapes are deployed. The shaft of one of the tape drums is threaded in which runs another threaded shaft. As the tape drum rotates, the shaft extends until it actuates a limit switch when the sampler has reached the limit of its travel. This causes a relay to latch reversing the polarity of the current to the motor. This in turn reverses the motor and the direction of rotation of the canted feed roller carriage causing the sampler to travel back up the tube. The wiring schematic diagram is shown in figure 4-5.

To summarize the design of the vertically deployed conical abrasive sieve sampler, a step by step sequence of events is listed in Table 4-I. The mechanics of providing the predetermined time intervals have not been incorporated in the prototype design, but could easily be done by the addition of a simple cam actuated timer.

TABLE 4-I

SEQUENCE OF OPERATIONS

Vertically Deployed Conical Abrasive Sieve Sampler Prototype

1. Sampler sequence is command initiated by turning on power to the motor.
2. The sampler travels down the support tube until the sampling head contacts the surface. If no surface contact is made the sampler travels to the limit of its stroke at which point a limit switch activates a relay to reverse polarity of the current to the motor thereby reversing it. This causes the sampler to return to the stowed position.
3. If the surface is contacted, the sampler continues to collect for a predetermined time interval or until the limit switch is reached. When either of these events occur, the current polarity is reversed as in step 2 and the sampler returns to its stowed position.
4. When the sampler reaches the stowed position, the motor carriage travel is stopped by the end plate of the tube. The canted feed roller carriage continues to advance overriding the spring preload until the friction clutch engages causing the sampling head to spin at a high speed transferring the soil to the collector.
5. The high speed spin is maintained for a predetermined time interval agitating the collector and transporting the soil to the delivery chutes. At the end of this interval the power is turned off which recycles the sampler to its original start position from which another cycle can be initiated if desired.

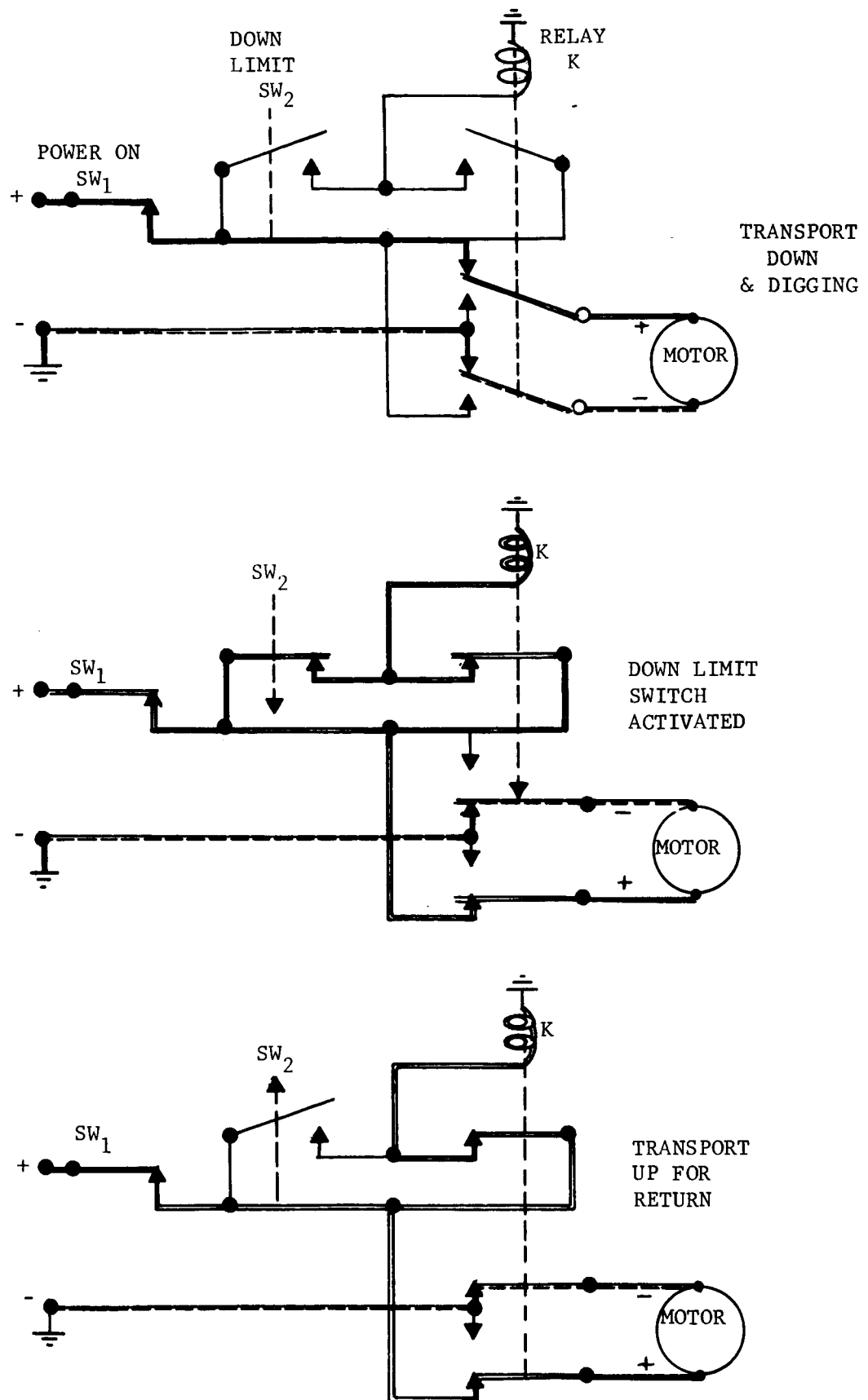


FIGURE 4-5 WIRING SCHEMATIC - CONICAL ABRADING SIEVE SAMPLER PROTOTYPE

The vertically deployed conical abrading sieve sampler prototype weight is 7.35 pounds. A weight breakdown by subassemblies is given in Table 4-II. It is estimated that two to three pounds could be removed from the existing design by carefully optimizing the various components.

TABLE 4-II

VERTICALLY DEPLOYED CONICAL ABRASIVE SIEVE
SAMPLER PROTOTYPE WEIGHT SUMMARY

<u>Component</u>	<u>Weight</u>
Motor	.72
Motor Carriage and Tape Take-up Mechanism	1.99
Sampling Head & Canted Feed Roller Assy.	1.15
Support Tube	1.84
Soil Collector Assy.	1.65
Total	7.35

4.1.2 FABRICATION

The completed subassemblies of this sampler are shown in figure 4-6. The assembly on the left is the sampling unit which consists of the sampling head, the drive shaft, the canted feed roller carriage, the motor carriage, and the tape take-up mechanism. Three different conical abrading heads are shown. The one mounted in the sampling head has two carbide blades brazed onto the cone adjacent to a radial slot. These are located 180 degrees apart. Adjacent to the sampling head are two other cutters. One of these has .125 diameter holes distributed over the entire surface. The other has .062 diameter holes located in 4 radial lines 90 degrees apart. The latter cone was also machined to a thickness of .005 inches rather than the .032 inches used on the other cutters.

Next to the sampler is located the support tube in which the sampler runs. At the lower end is located the soil sampler collector or receiver. Figure 4-7 shows the sampler in the support tube in a partially extended position.

The most complicated assembly is the gearbox located on the end of the motor carriage which converts the motor output to a two speed coaxial output. A two stage planetary gear drive with an overall ratio of 17.5:1 reduces the motor output of 375 rpm to 21.5 rpm. On the initial assembly it was discovered that the rotation of the high speed spin shaft was in the wrong direction for the overrunning clutch to release. An additional gear train

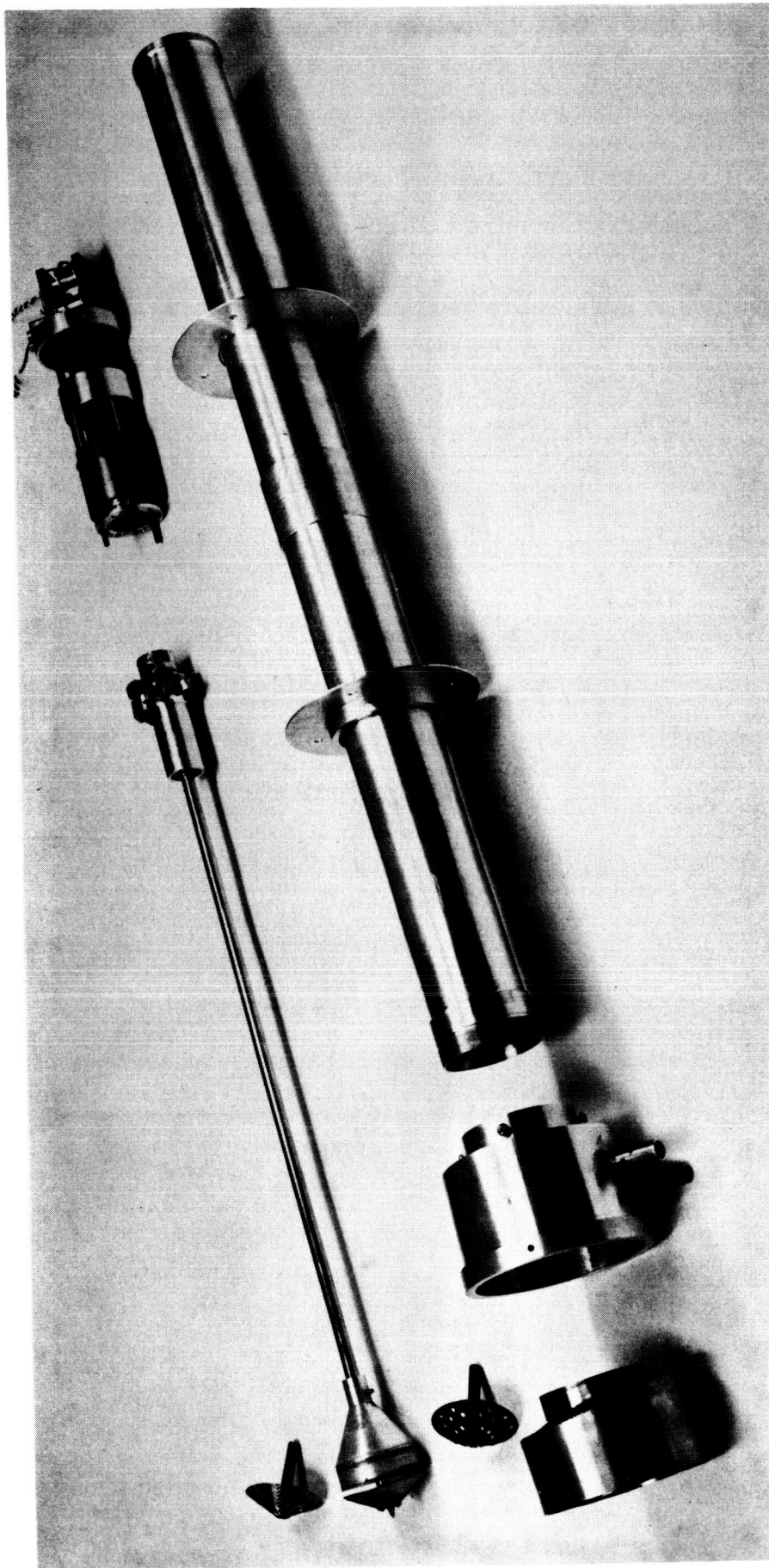


FIGURE 4-6 CONICAL ABRASIVE SIEVE SUBASSEMBLIES

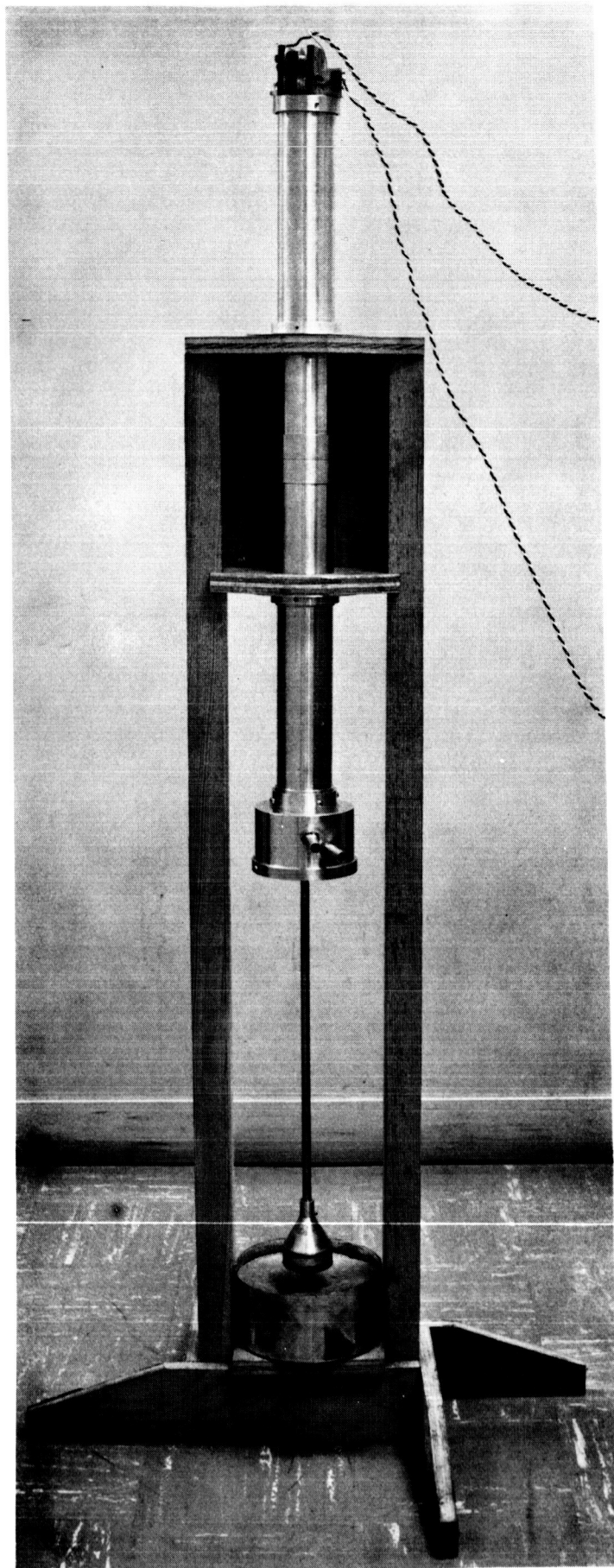


FIGURE 4-7 VERTICALLY DEPLOYED CONICAL ABRASIVE SIEVE SAMPLER ASSEMBLY

was incorporated to reverse the rotation of the high speed spin shaft. The overall ratio of this gear train is 1.27:1 resulting in a final output spin of 475 rpm of the high speed output shaft. The overrunning roller clutches are commercially available item which performed very reliably. The size of the clutch used in this sampler is very small. The clutch fits on a .25 diameter shaft and runs inside a .625 diameter housing. The rated torque capability of this sampler is approximately 20 inch pounds with a break away torque of 80 inch pounds. In order to reduce the load on each clutch, two clutches in parallel were used. Initially, these clutches ran in an aluminum housing. During an early run the clutches froze in the housing. On disassembly it was discovered that a thin wall produced by external machine cuts had broken through. A hardened steel insert was pressed into the canted feed roller carriage to form the clutch housing eliminating this problem.

The motor carriage is supported in the tube by eight rollers. Each roller is a miniature ball bearing mounted between two beryllium/copper springs which preload the roller against the tube wall. Each pair of springs are supported at the center allowing a roller to be cantilevered in both directions from the supporting ring. No problems were encountered in this assembly. The motor carriage moves freely along the axis of the tube with no tendency to slip or rotate when a torsional load is applied. Some slight brinelling of the aluminum support tube was noted at the edges of the rollers; however, this was not severe enough to cause any problems in operation.

The major problem area encountered in fabrication and assembly was encountered in achieving the requisite end fixity and adjustment of the canted feed roller torsion bars. The original design relied on a press fit at either end of the .093 diameter beryllium/copper torsion bars. This proved to be inadequate and did not allow for adjustment of the canted roller position. It became apparent that this adjustment was fairly critical. If too little preload was achieved on the canted roller, axial slippage occurred at less than the desired thrust. If too much preload or unequal preload was achieved, either torsion bar breakage or yielding at the support points occurred. The torsion bar failure occurred as shown in figure 4-8. The failure can in part be attributed to the very small fillet radius initially used. In order to increase the traction of the canted feed roller for a given preload, the original ball bearing roller was replaced with a steel roller machined with a slight crown. This crown was knurled with a medium diamond knurl to provide a tread. The results of this change produced the requisite traction although the tread on the roller embosses the surface of the aluminum tube. The fixed end of the torsion bar was redesigned so that radial adjustment could be made by shimming and the cant angle could be adjusted continuously from zero to 15 degrees. In order to assess the relative importance of roller preload and cant angle on the operation of the sampler feed mechanism, a series of runs were made to measure the axial thrust developed and the unrestricted rate of travel. The data collected are listed in Table 4- III. These characteristics are shown in the curves in figure 4-9.

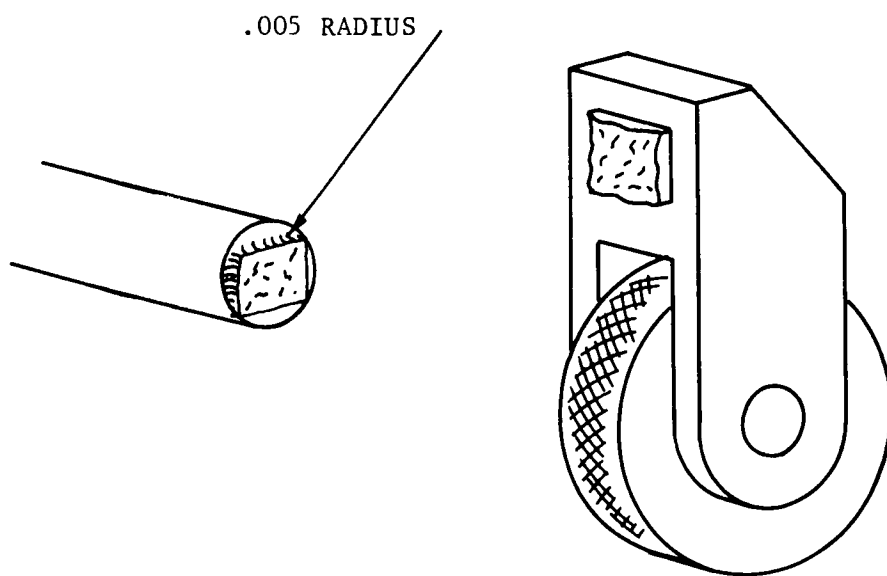


FIGURE 4-8 TORSION BAR FAILURE MODE

TABLE 4-III

MECHANICAL CHARACTERISTICS OF VERTICAL SAMPLER FEED MECHANISM

<u>Run Lbs</u>	<u>Roller Angle</u>	<u>Roller Preload</u>	<u>Force Lbs</u>	<u>Time/ 6" Travel</u>	<u>Inches/ Min</u>
1	5°	.027	15.4	17.5 sec	21.2
2	5		15.2		
3	5		15.1		
4	5		15.0		
5	10°	.027	18.25	13.5 sec	26.7
6	10		18.75		
7	10		18.75		
8	10		18.75		
9	15°	.027	20.25	9.4 sec	38.3
10	15		19.75		
11	15		19.75		
12	15		19.50		
13	5°	.032	13.75	32.7 sec	11.0
14	5		12.00		
15	5		12.25		
16	5		12.50		
17	10°	.032	19.75	16.0 sec	22.5
18	10		20.25		
19	10		20.00		
20	10		20.50		
21	15°	.032	21.75	7.5 sec	48.9
22	15		20.50		
23	15		21.75		
24	15		21.75		

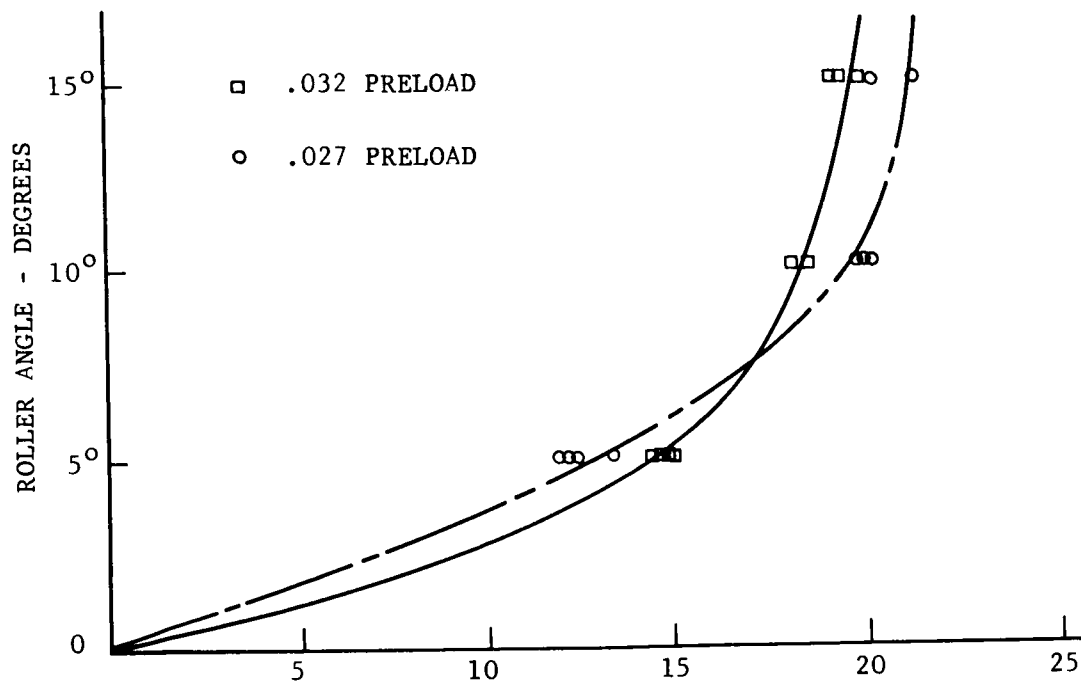
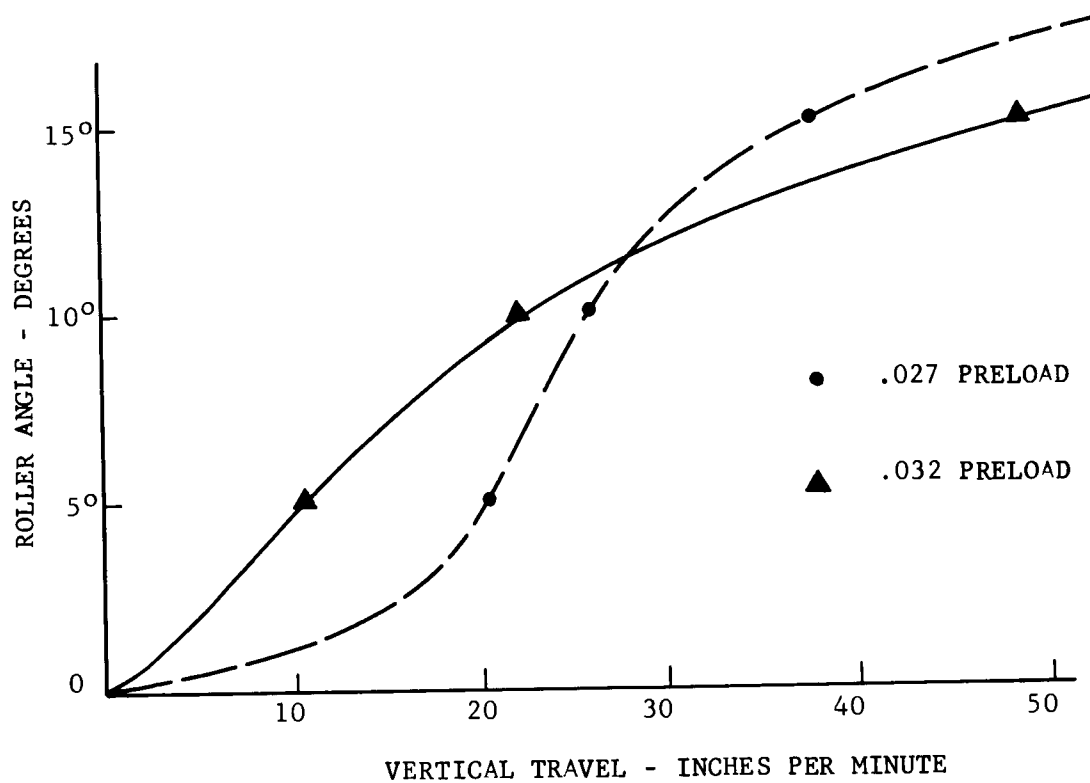


FIGURE 4-9 VERTICAL DEPLOYED SAMPLER FEED CHARACTERISTICS

It is seen that increasing the preload actually reduced performance for the smaller roller cant angles. This is probably due to the nonlinear deflection characteristics of the torsion bar under combined loading and cross coupling of deflections between the bending mode and torsional mode. The roller cant angle of 5 degrees with a roller preload deflection of .027 inches was selected as the operational adjustment to be used in the evaluation testing. With these adjustments an axial thrust of 15 pounds and an advance rate of 20 inches per minute is achieved which were considered to be sufficiently close to the initial design objectives.

Four conical abrading sieves or cutters were fabricated as listed in Table 4-IV.

TABLE 4-IV
CONICAL ABRADING SIEVE TYPES

Type	Description
1	.062 wide radial slot 180° apart. Carbide blades onto trailing edge of slot.
2	.072 diameter holes arranged in 4 radial lines 90° apart. Trailing edge upset outward and leading edge upset inward.
3	.125 diameter holes arranged in a random pattern over the surface of the cutter. Trailing edge upset outward.
4	Tungsten carbide grit brazed over surface of cone with random patterns of .040 diameter holes.

Of these four conical abrading sieve or cutter types, only 1 and 2 were evaluated in the test program discussed in paragraph 4.1.3. Cutter type 3 was machined from 4130 steel and heat treated. This cutter was found to be deficient in two respects. First, the hole size was much too large and allowed the sample to run out during retraction. Second, the hole pattern did not provide a cutting edge over the entire surface, particularly near the center. As a consequence, a bearing surface is established and all further advance of the abrading head is halted in cemented soils such as hardpan.

Cutter type 4 was machined from stainless steel. In the attempt to braze the tungsten carbide grit to the stainless steel, a low melting point eutectic alloy between the braze material and the stainless steel was formed resulting in a badly deformed and cracked part. Although other methods and materials are available with which to fabricate such a part, no further attempts were made to fabricate this cutter. No difficulty was encountered in the fabrication of cutter type 1 except that the cutting edge was not carried sufficiently close to the center or the outer edge. This again

left bearing surfaces which could halt the advance into cemented soil. The carbide blades were extended to the inner edge of the cavity provided by the tip auger to eliminate this bearing surface. In preliminary runs in vesicular pumice, sufficient torque was generated to cause the torque tube to slip on its shaft to which it was attached by a collet type of chuck. The torque tube shaft was modified so that it was pinned to the drive shaft to prevent slippage. The height of the carbide blade was also ground down to provide a cutting edge .032 inches deep to reduce the amount of material being cut in a single pass.

Cutter type 2 was machined from 4130 steel and heat treated. The hole patterns were located on 4 radial lines 90 degrees apart. Each alternate set of holes were staggered with respect to the other in order to provide a continuous cutting surface from the center to the edge of the cutter. The thickness of the conical surface for this cutter was reduced from the .032 inches used on the other types to a thickness of .005 inches. The intent was to provide some flexibility to allow the cutter to conform somewhat to the surface being sampled. Also, the thinner material at the holes was felt to provide an easier entry by the soil particles which had been abraded from the surface. The four cutters are shown in figure 4-10.

4.1.3 PERFORMANCE EVALUATION

The performance evaluation testing of the vertically deployed conical abrading sieve prototype was accomplished using cutter types 1 and 2 previously described. Thirty eight test runs were completed as defined in the test matrix shown in figure 4-11.

The first eight runs were made to evaluate the effect of slit width opening on performance. The last two runs were made to evaluate the need for spring loading the centering tip. The first four runs in compacted silt produced twice the volume of sample at 4 mm than it did at 2 mm slit width opening. Since the enclosed volume did not double, the narrower slit width prevents the sampler cavity from completely filling during sampling. No appreciable differences were noted in the effect of slit width in the sand model. Collection of sand for both sampler types was poor due to sand running out of the slots and holes during retraction of the sampler. Either smaller openings or some sort of spring valve will be needed to retain the sample in the sampler. In the case of cutter type 2, locating the abrading holes at 180 degree radial lines rather than 90 degrees would also improve its performance in sand.

The initial radial clearance between the sampler hood and the tube, in which the drive mechanism runs, was .005 inches. In these early tests, binding between the hood and the tube occurred due to entrapped soil particles. The diameter of the cutter head was reduced by .060 inches.

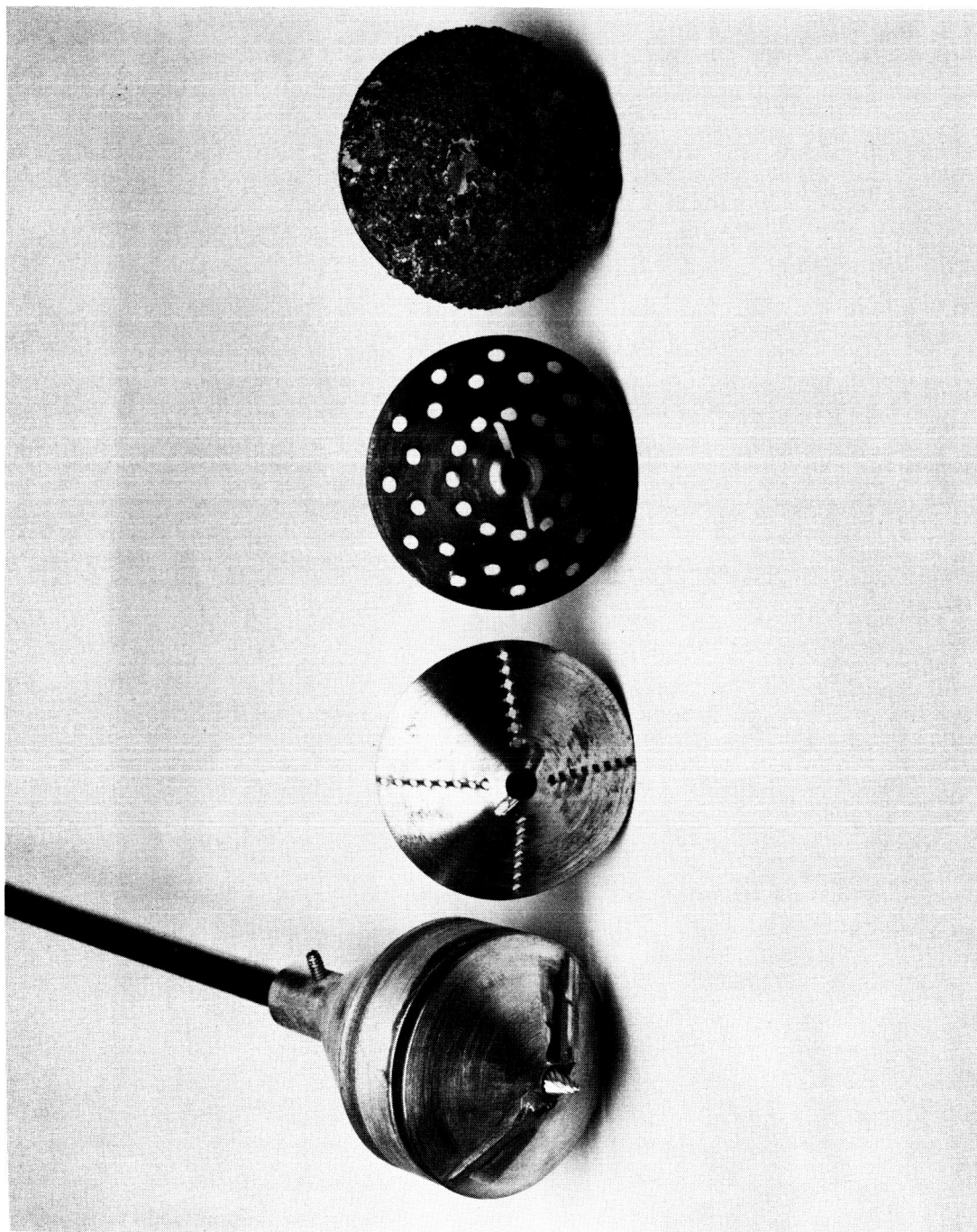


FIGURE 4-10 CONICAL ABRASIVE SIEVE CUTTER TYPES

This not only eliminated the binding but allowed the head to vibrate or bounce off the tube wall due to the unbalance caused by the loose weight in the hood. In the case of silt which is quite heavily compacted in the sampler head, this vibration greatly reduced the time required to spin dump the sample. A test run with a slit opening of 1.375 inches was made in silt. A total of 44.5 grams of soil was collected in 15 seconds of actual digging time. Thus, the hood design actually limits the total size of sample which could be collected. This suggests a hood design which provides a larger volume above the slit for soil to collect in.

All subsequent test runs were made using a slit width of 4 mm and a fixed centering tip except for runs 37 and 38 which used a spring loaded tip. The effect of the spring loaded tip is important only when sampling a hard cohesive surface such as hardpan. It allows the cutting edges to reach the surface more rapidly thereby wasting less time. A probable design improvement would be to provide a very short fixed centering tip since it does not appear to be needed except on hard surfaces. It was noted that sampling on the hardpan surface produced an equivalent amount of loose soil around the hole as was collected by the sampler.

The average drilling power required is summarized in figure 4-12 which is seen to vary from about 9 watts on hardpan to 13 watts in rubble. The slightly higher power required in the particulate soil models is probably due to the friction load acting on the sampler head and hood since the sampler buries itself rather quickly in these models. No significant difference in the power requirements for cutter types was noted.

At periodic intervals during the testing, the drive mechanism was taken from the tube and measured across the rollers in order to detect wear or permanent deformations. No significant changes were noted. The tube itself did show significant changes in that the knurled rollers produced considerable texturing of the surface, particularly where advancement was halted by the surface being sampled. During test run number 17, it was noted that aluminum flakes fell out of the tube onto the surface. This was not noted on the preceding runs in which the sampler advance into the soil model was more rapid. In runs 17 through 20, this flaking continued and a plot of hole diameter made in the hardpan by the sampler showed decreasing size in successive runs. This is shown in figure 4-13. This indicates that tube wear is much more rapid on the hardpan model than in the particulate models. This increased wear is due to the fact that the feed rate is so low, the canted rollers run over the same surface repeatedly. This does not occur in the particulate soils such as sand or silt. The use of other tube materials such as steel or fiberglass can probably eliminate or reduce this effect.

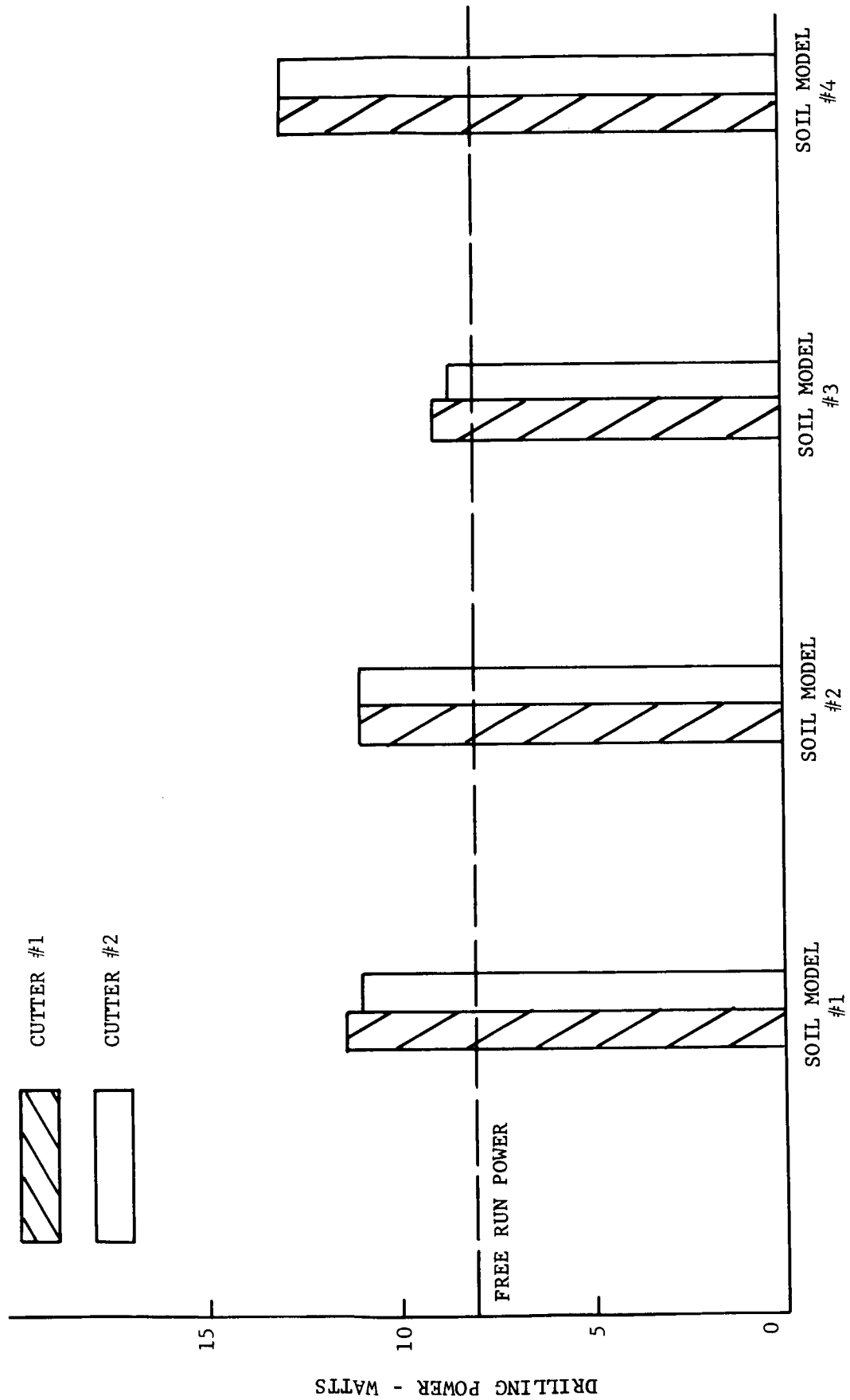


FIGURE 4-12 POWER REQUIREMENTS FOR VERTICALLY DEPLOYED SAMPLER

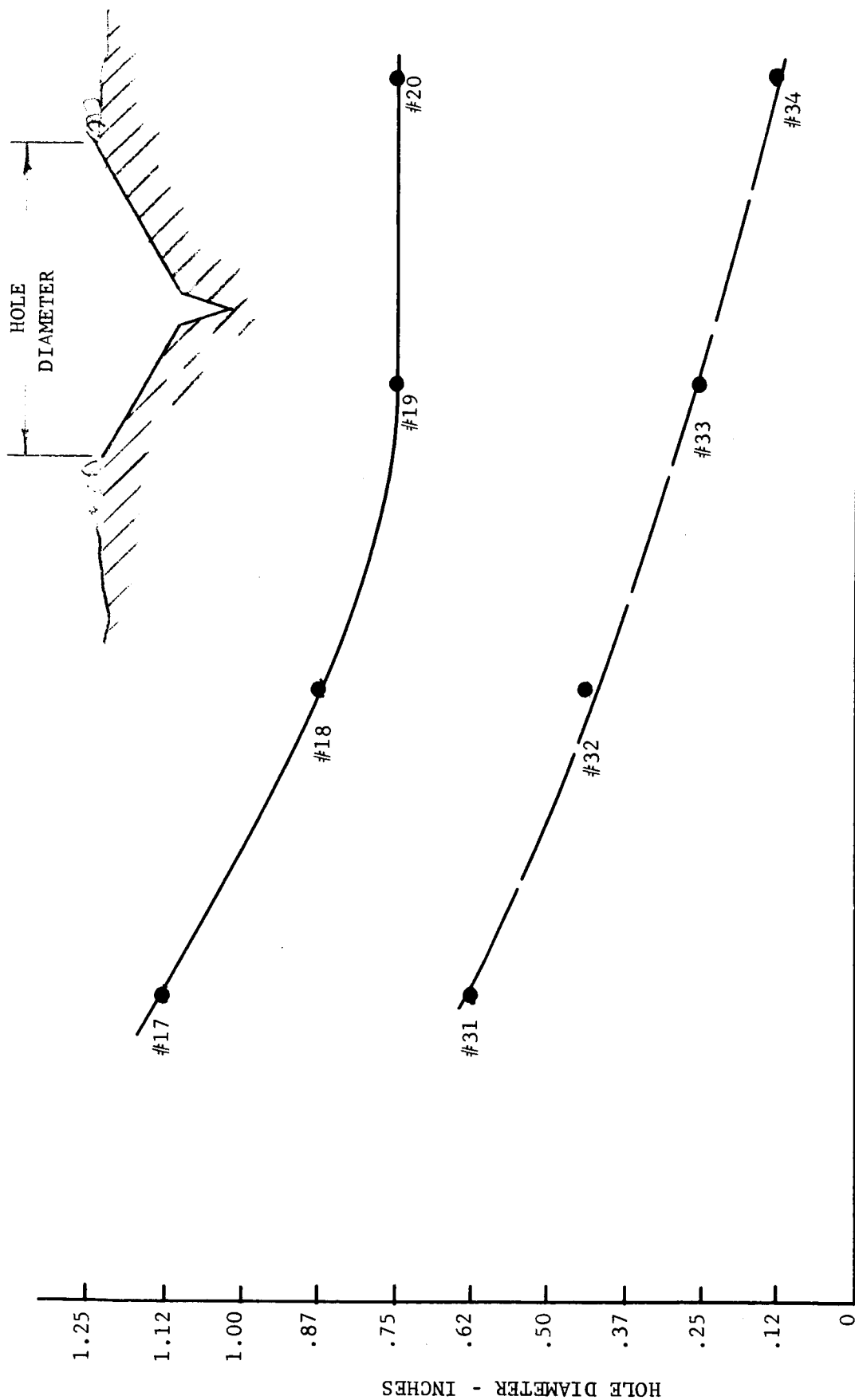


FIGURE 4-13 HOLE SIZE VARIATION AFTER SUCCESSIVE RUNS IN HARDPAN

After the completion of test series number 20, the canted roller head diameter was checked and found to be 2.805 inches, which was what it was prior to the start of test run 9. The angle of the rollers was also found to be within tolerances. The problem of breaking torsion bars appeared to have been solved by increasing the fillet radius at the shoulder of the torsion bar where it mates with the roller mount.

A visual inspection of the inside of the tube showed heavy brinelling in the areas where the canted rollers acted during the first 23 complete digging cycles. A check was made of the diameters of this tube every inch over its length. These measurements were made using "Intramic's," three point micrometers. The largest diameter difference was found to be .001, however, it is felt that the three point micrometer will average local small variations such as those caused by the knurled canted rollers. A check of force available in those localized areas showed an available force of 8 to 10 pounds as compared to the original 15 pounds before the tube was worn.

The tube was replaced after run number 20 to complete the test matrix. Runs 21 and 22 were actually performed last with runs 35 and 6. This was done in order to collect a maximum amount of data before attempting to sample the rubble model since this model was felt to be most hazardous to the sampler.

The test results obtained in the rubble models were most interesting. The sample was prepared as indicated in Table 4-V. For each sampler cutter type, the sampler was defeated once and succeeded once. In one case the sampler came down on a boulder or rock and subsequent progress was stopped. In another case the sampler came down on the edge of a rock forcing it down. This allowed the sampler head to deflect to the side enough to enter the rubble. This sequence is shown in figure 4-14. On entering the rubble the rock was pushed aside and another large rock fell in behind the hood. The two rocks acting as a toggle trapped the sampler preventing its return. A sample was collected but not delivered.

An interesting characteristic of the sample collected in the rubble model is that more sand than silt was collected although equal parts of each by weight were mixed together uniformly. The detailed compositions of the soil models are given in Table 4-VI. The soil particle size analysis for the collected sample and residual abraded soil around the hole is given in Table 4-VII for cutter no. 1 and Table 4-VIII for cutter no. 2. No interpretation as to why more sand was collected is attempted.

The final results of the testing is presented in figure 4-15 in terms of soil collection rate for the various soil models and cutter types. The soil collection rate was determined by dividing the total sample collected by the actual digging time. This time varied with the model used. The digging time for the hardpan was 5 minutes in each case. The dig time for the other models was determined when a penetration into the soil of 2.625 inches was achieved.

TABLE 4-V

SOIL MODEL PREPARATION

Model No. 1 - Compacted Silt	
Composition:	Compacted crushed olivine basalt, grain size <325 mesh (<44 μ); grain shape very angular, cohesion obtained by packing and grain interlocking.
Preparation:	Soil model prepared in a 1-gal tin can; cohesion obtained by tapping on a solid surface 25 times from a height of approximately 1-inch.
Model No. 2 - Cohesionless Sand	
Composition:	<p>Cohesionless Nevada No. 60 natural dune sand, grain size range as follows (Percent by weight):</p> <p>0.6% retained on a No. 30 standard sieve (595μ)</p> <p>7.7% " " " No. 40 (420μ)</p> <p>18.3% " " " No. 50 (297μ)</p> <p>29.0% " " " No. 70 (210μ)</p> <p>33.7% " " " No. 100 (149μ)</p> <p>9.0% " " " No. 140 (105μ)</p> <p>1.77% through No. 140 sieve (<100μ)</p> <p>Grain size range: 91.6% <300μ and 89.3% >150μ;</p> <p>Mean grain size: 250μ (60 mesh)</p>
Preparation:	Soil model prepared in a 1-gal tin can; prior to initial test and between tests loose packing was achieved by rolling the can through at least four complete revolutions, carefully righting it, and carefully raking the sand surface flat.
Model No. 3 - Hardpan	
Composition:	<p>Simulated cemented hardpan, mised as follows (materials in percent by weight):</p> <p>44.6% Nevada No. 60 sand, as above (Model No. 2)</p> <p>42.4% Ottawa No. 398 silt, grain size 100% <44μ</p> <p>11.9% Common cement</p> <p>1.1% Hematite silt dye, grain size 100% <44μ</p>
Preparation:	The bulk dry materials were completely mixed, water was added until a consistence of bulk cement was obtained, then one gallon tin cans were filled one-half full with wet mixture. Model was allowed to harden for 2 months.
Model No. 4 - Rubble	
Composition:	<p>Randomly mixed rubble composed of the following materials (percent by weight):</p> <p>25% Natural olivine basalt cobbles, sized to 12.5 cm</p> <p>25% Crushed olivine basalt pebbles, sized between 1.25 and 2.5 cm</p> <p>25% Crushed olivine basalt silt, as above (Model No. 1)</p>
Preparation:	Rubble mixture was prepared in a 7 x 7 x 7 inch box with a lid. Between tests rubble was randomized by rotating box through 90° four times with the box turned 90° prior to each rotation; a final tap from a height of approximately 1 inch as the box was returned upright served to level the surface; and each sample was taken from the exact center of the box.

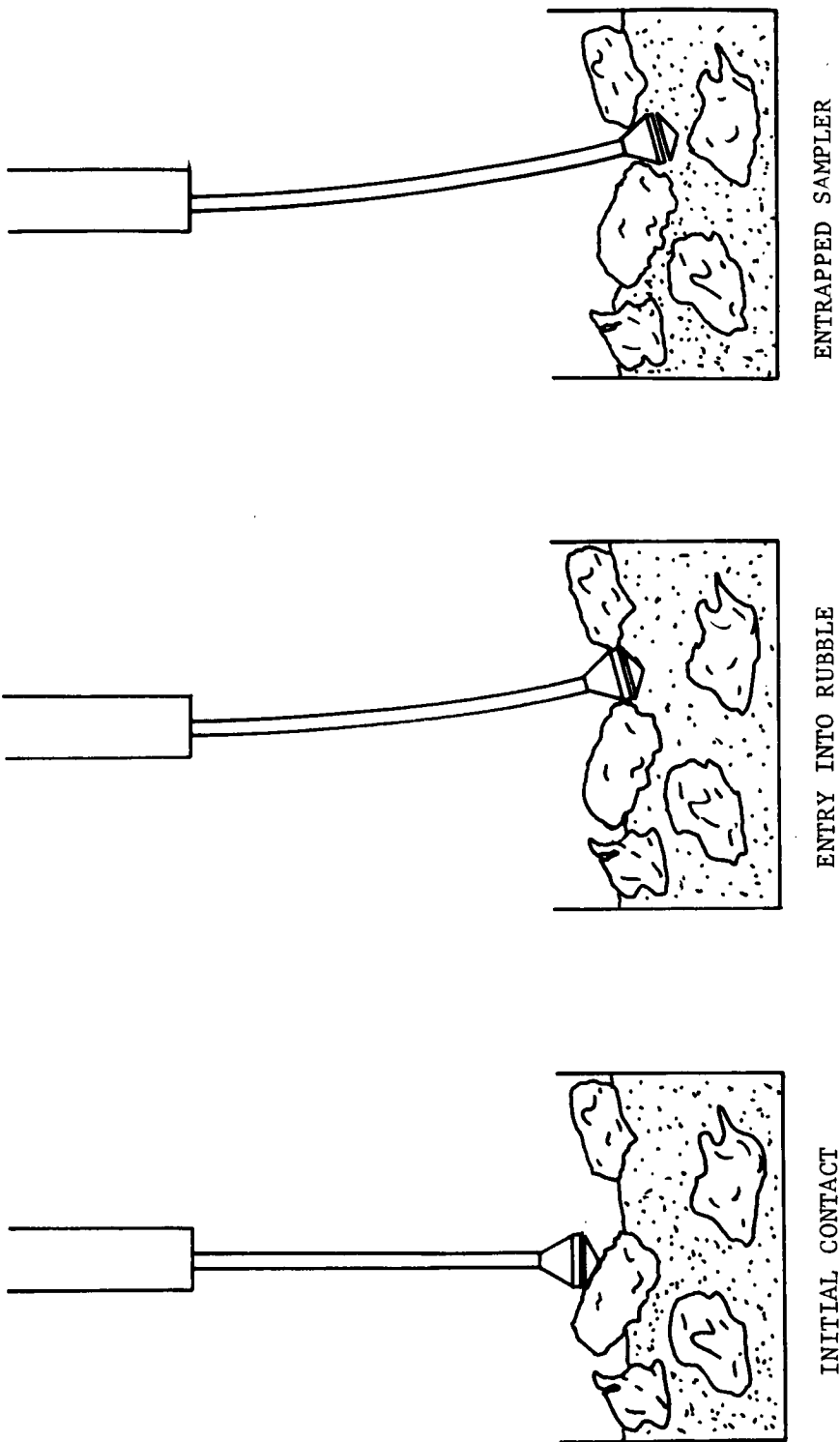


FIGURE 4-14 SAMPLER ENTRAPMENT IN RUBBLE

TABLE 4-VI

PARTICLE SIZE DISTRIBUTION ANALYSIS

Sieve Analysis of Material Collected From Tests Conducted on Soil Model No. 3
(Hardpan)

Summary of Hardpan Tests		Weight of Soil (in grams)	
Test No.	Cutter Head Type	Spin Dumped	Residual Soil Around Hole
17	1	1.50	2.35
18	1	0.60	1.60
19	1	0.35	1.05
20	1	0.35	0.90
37	1	5.60	4.40
38	1	3.25	2.95
Total		11.65	13.25

Sieve Analysis of Material Collected From Tests Conducted on Soil Model No. 4
(Rubble)

Cutter No. Test No.	1 21	1 22	2 36
Fraction recovered (by weight in grams)			
Basalt chips	Trace	Trace	None
Sand (Nevada No. 60)	24.80	22.90	22.15
Silt (Crushed basalt)	12.80	17.50	11.10
Lost in sieving	0.40	0.45	0.15
Total collected	38.00	40.40	32.40

TABLE 4-VII

SIZE ANALYSIS OF SOIL FOR CUTTER NO. 1

Spin Dumped Sample

Weight of Soil (in grams)	U.S. Standard Sieve	Particle Size
0.04	Retained on No. 10	>2 mm
0.02	No. 18	1 mm
0.06	No. 35	500 μ
1.42	No. 60	250 μ
2.87	No. 120	250 μ
1.28	No. 230	63 μ
3.30	No. 325	44 μ
2.56	Through No. 325	<44 μ
0.10	Lost in Sieving	
Total 11.65		

Residual Soil Around Hole

Weight of Soil (in grams)	U.S. Standard Sieve	Particle Size
0.15	Retained on No. 10	>2mm
0.30	No. 18	1 mm
0.57	No. 35	500 μ
2.52	No. 60	250 μ
4.59	No. 120	125 μ
1.45	No. 230	63 μ
0.67	No. 325	44 μ
2.95	Through No. 325	44 μ
0.05	Lost in Sieving	
Total 13.25		

TABLE 4-VIII

SIZE ANALYSIS OF SOIL FOR CUTTER NO. 2

Spin Dumped Sample

Trace (2 particles)	Retained on No. 18	>1mm
0.07	No. 35	500μ
0.55	No. 60	250μ
1.31	No. 120	125μ
0.48	No. 230	63μ
0.44	No. 325	44μ
0.94	Through No. 325	<44μ
0.05	Lost in Sieving	
Total 3.85		

Residual Soil Around Hole

Trace (2 particles)	Retained on No. 10	>2mm
Trace (12 particles)	No. 18	1mm
0.05	No. 35	500μ
0.08	No. 60	250μ
0.41	No. 120	125μ
0.18	No. 230	63μ
0.03	No. 325	44μ
0.26	Through No. 325	44μ
0.04	Lost in Sieving	
Total 1.05		

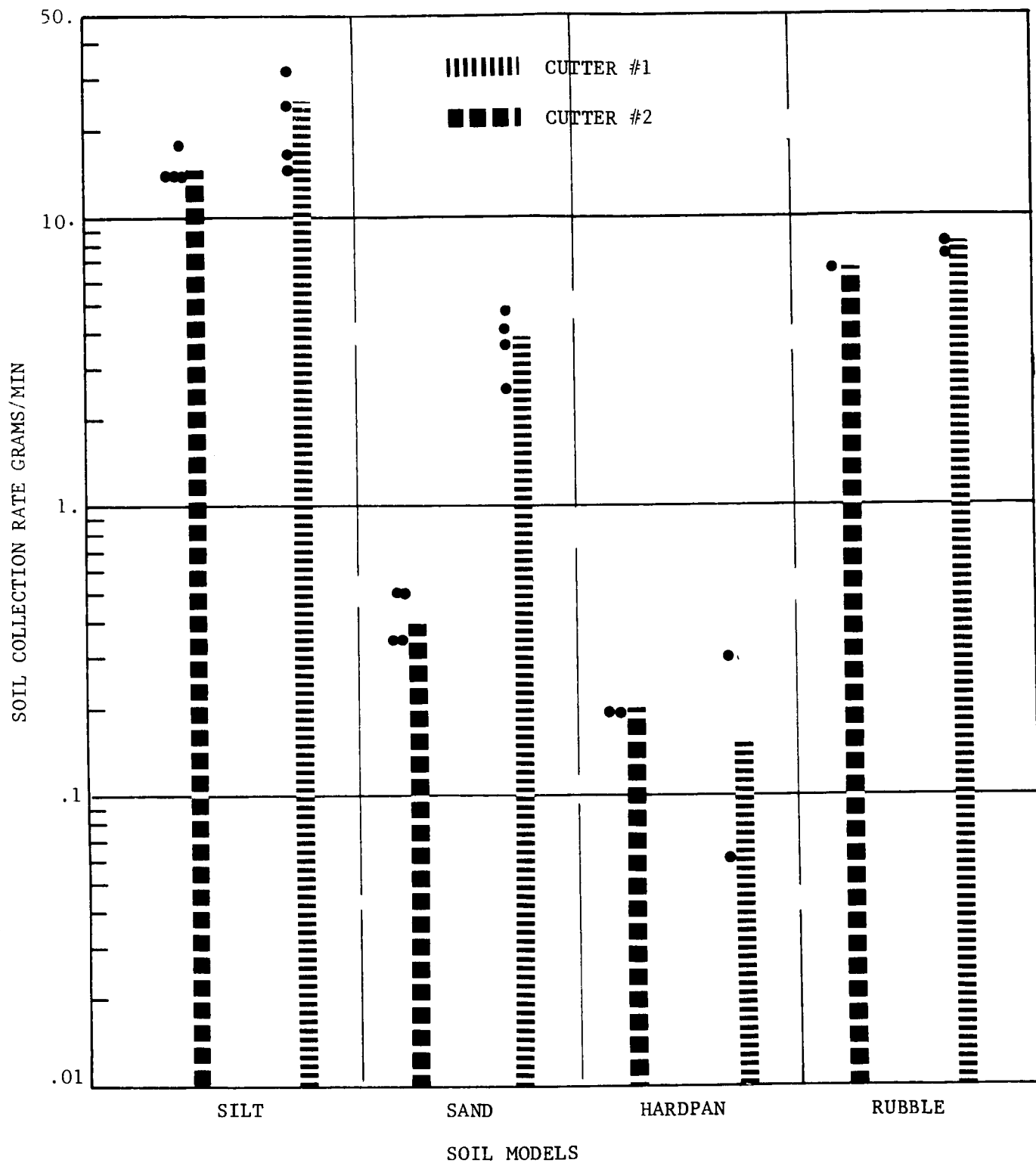


FIGURE 4-15 SOIL SAMPLE COLLECTION SUMMARY VERTICALLY DEPLOYED ABRADING SIEVE

It is seen from figure 4-15 that the highest collection rate was achieved in model 1 (silt) and model 4 (rubble). The sand (model 2) was low because most of the sand ran out of the sampler head on the return trip. The hardpan (model 3) is low because of the high resistance to abrasion encountered. The largest total samples of about 25 grams were achieved in the silt and rubble models. The sample size for hardpan was in the order of a gram. If the sampler heads are modified to retain the sand, collection should be as high as for silt.

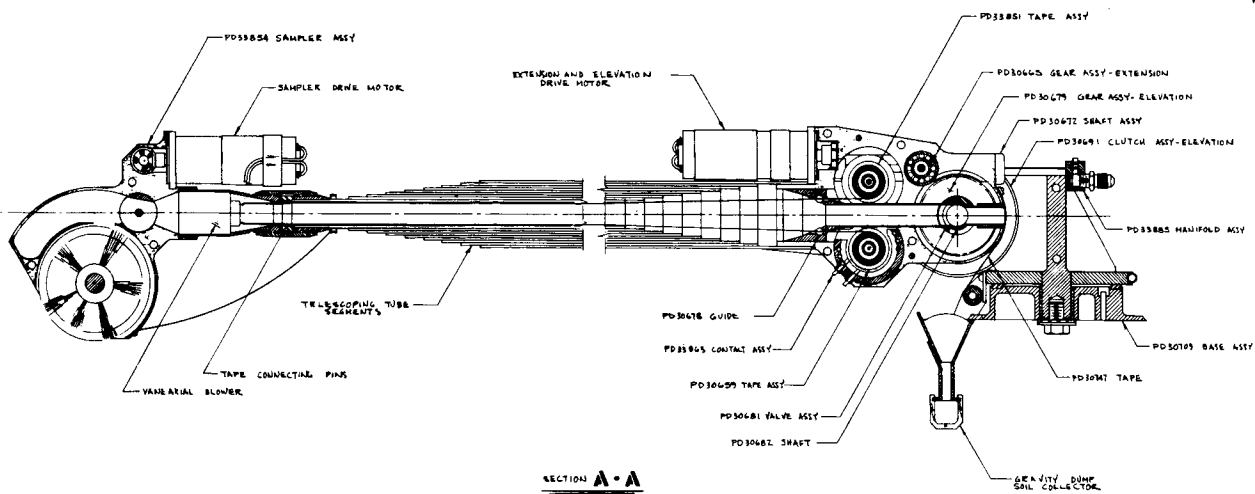
Another interesting result of grading the abraded material is that the hardpan particle size distribution achieved by abrasion was the same as the particle size distribution used in the constituents before bonding and curing the model.

4.2 HORIZONTALLY DEPLOYED ROTATING WIRE BRUSH/ TELESCOPING BOOM PROTOTYPE

4.2.1 DESIGN

This sampler prototype is inherently a more sophisticated and hence complicated design. The basic requirement for this sampler was to have the capability of being deployed over the surface in a horizontal direction from the payload and sampling at various points on the surface. The actual design evolution was supported by breadboard testing of various components of the system. These breadboard tests are discussed in Section 3.0. This sampler prototype design can be broken down into two distinct parts or assemblies. One is the rotating wire brush sampling head and the other is the telescoping boom and support assembly. The complete assembly is shown in figure 4-16 and figure 4-17. In order to develop the design approach used for this sampler, the operational sequence is described step by step in Table 4-IX. This sequence is shown schematically in figure 4-18. It should be noted that the schematic diagram and operational sequence do not include a snubber system incorporated into the engineering prototype in order to reduce the load on the elevation gear train. The snubber system would not be required on the flight hardware since the reduced gravity of Mars would not overload the gear train.

At this point, it is worthwhile to note that the type of boom used with this sampler does not necessarily have to be a pneumatically deployed telescoping boom. During the conceptual design phase, mechanically furlable booms were investigated in detail parametrically and appear to offer some advantages as was discussed in Section 2.3.1.



4-32-1

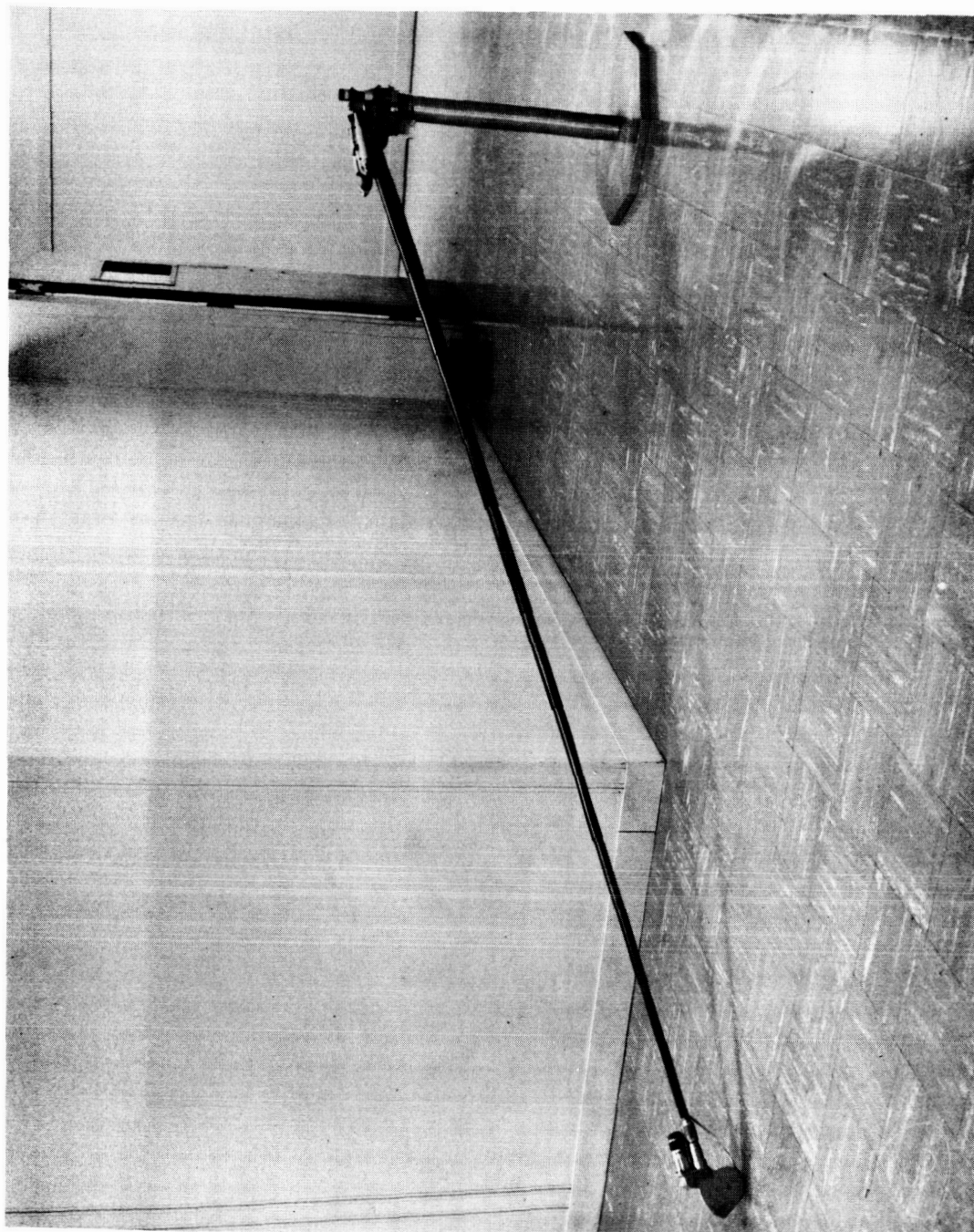
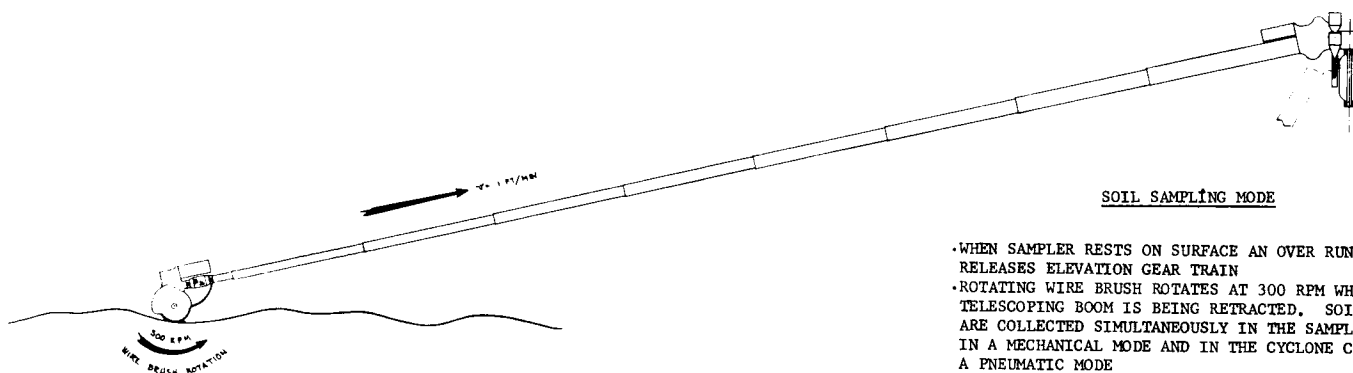


FIGURE 4-17 ROTATING WIRE BRUSH/TELESCOPING BOOM PROTOTYPE SAMPLER

TABLE 4-IX

ROTATING WIRE BRUSH/TELESCOPING BOOM SAMPLER
PROTOTYPE OPERATING SEQUENCE

1. On command the power is turned on and pneumatic pressure is applied to the boom. The rotation of the tape deployment gear train closes a valve at the base of the boom and the rotation of the wire brush gear train closes a valve at the tip of the boom. This seals the boom allowing it to be pressurized.
2. The appropriate gear train is engaged by means of an overrunning clutch to feed the tapes at the proper rate to control boom deployment. Full deployment is achieved in approximately a half a minute.
3. While the boom is extending, another overrunning clutch engages the elevation gear train causing the boom to be erected to a vertical position.
4. When extension and erection are complete, a switch driven through a gear train from a tape deployment drum is actuated reversing the current polarity to the motors. Reversal of the motor rotation opens the valves at each end of the boom allowing a pneumatic flow to be established through the sampler and boom into a cyclone collector.
5. Retraction of the boom is initiated at the rate of one foot per minute and the sampling head lowered to the surface at a controlled rate by the elevation gear train. When the surface is contacted, the elevation gear train is automatically released by the overrunning clutch.
6. The sampler is drawn over the surface along a radial line at one foot per minute with the wire brush rotating. Small or fine soil particles are transported continuously during this operation into the cyclone collector. Simultaneously, larger particles are collected in a soil collection chamber incorporated in the sampler head.
7. When retraction of the boom is completed, the polarity reversing switch is again actuated causing the initial erection cycle to be repeated. When the vertical position is reached and the valves reopen, the soil collected in the sampling head is dropped down the tube to be delivered in a gravity dump mode.
8. As the boom reaches the vertical position, it depresses a plunger which acts as the vertical stop. This in turn depresses a stop pin which will allow a spring drive to rotate the boom support base to a new azimuth position before the sampler is lowered to the surface again. Sampling cycles are repeated automatically until interrupted by command.



SOIL SAMPLING MODE

- WHEN SAMPLER RESTS ON SURFACE AN OVER RUNNING CLUTCH RELEASES ELEVATION GEAR TRAIN
- ROTATING WIRE BRUSH ROTATES AT 300 RPM WHILE THE TELESCOPING BOOM IS BEING RETRACTED. SOIL SAMPLES ARE COLLECTED SIMULTANEOUSLY IN THE SAMPLING HOOD IN A MECHANICAL MODE AND IN THE CYCLONE COLLECTOR IN A PNEUMATIC MODE
- THE TELESCOPING BOOM AND SAMPLER WEIGHT PROVIDE THE NORMAL FORCE KEEPING THE BRUSH IN CONTACT WITH THE SURFACE
- RETRACTION IS COMPLETED IN 10 MINUTES - AT THE END OF RETRACTION THE DEPLOYMENT DRIVE MOTOR IS AGAIN REVERSE ENGAGING THE ELEVATION DRIVE TRAIN WHICH ROTATES THE BOOM TO A VERTICAL POSITION DELIVERING THE MECHANICALLY COLLECTED SOIL SAMPLE BY GRAVITY DUMP
- ACHIEVING THE VERTICAL POSITION AGAIN MAY BE UTILIZED ACTUATE A RELEASE ALLOWING THE SUPPORT BASE SPRING TO DRIVE THE SAMPLER TO A NEW AZIMUTH SETTING FOR SUCCESSIVE RUNS OVER A FRESH SURFACE

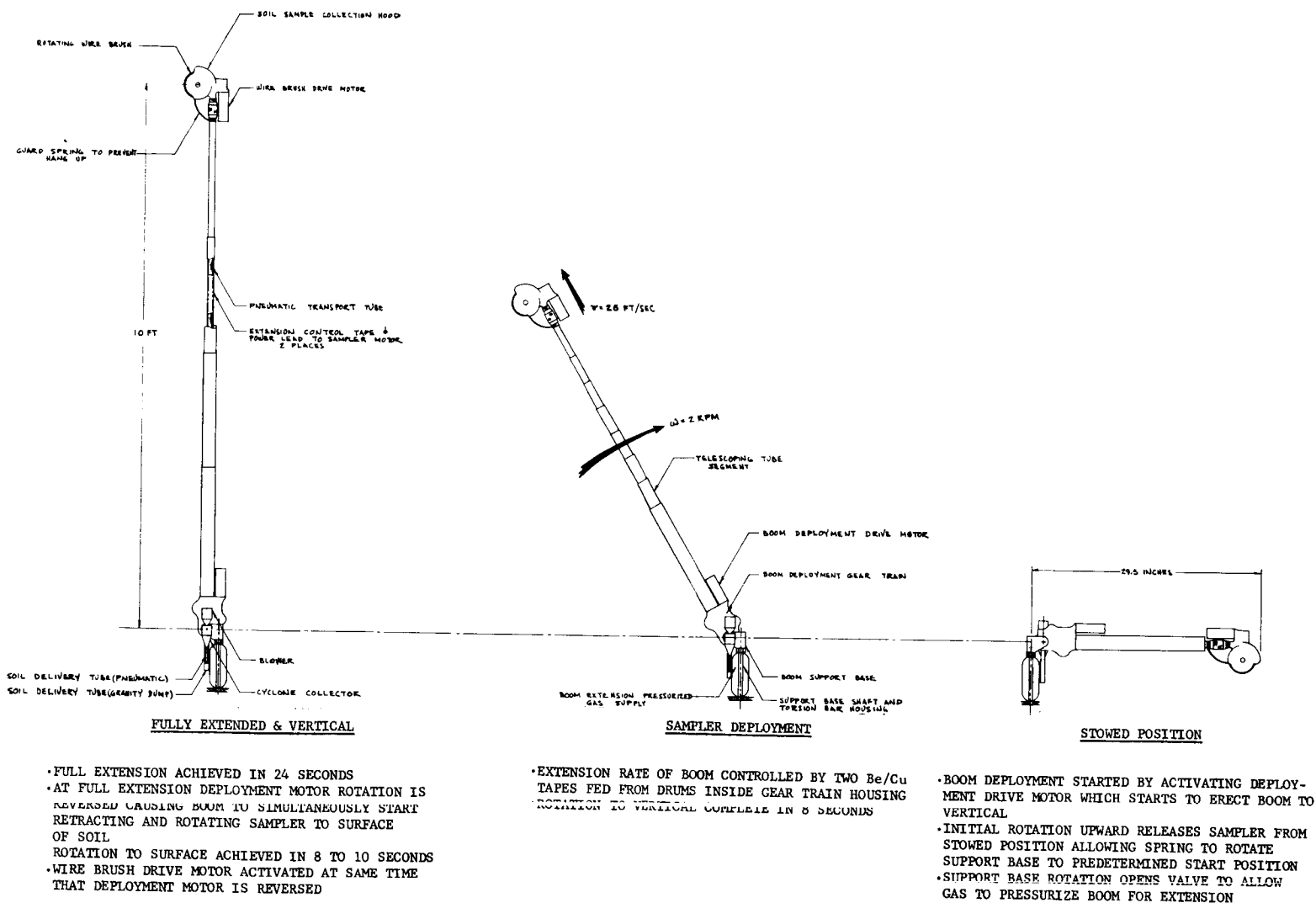


FIGURE 4-18 SCHEMATIC DIAGRAM - ROTATING WIRE BRUSH DEPLOYMENT

The decision was made that the furlable boom would pose a development problem which would be beyond the scope of this contract. Although a variety of furlable tapes have been made in lengths up to several hundred feet, they have been intended for space applications with very little structural strength requirements. These furlable tapes are open tube sections which are exceptionally weak in torsion. For an application such as a boom to support a soil sampler on Mars, the tape geometry will require the lowest permissible tube diameter to wall thickness ratio. The parametric studies show that the lowest ratio would be obtained with a fiberglass tube. Since such a tube has not been manufactured to date, it represented a fabrication development problem beyond the scope of this effort. A preliminary design concept for a double furlable tape in which one is folded inside the other was completed and is included in appendix C as drawing number PD25179. The fundamental idea behind the double furlable boom was to achieve some degree of torsional strength through the shearing friction forces existing between the tape surfaces where they contact each other.

To further explore the possibilities of using a boom which had already been developed, a trip was made to General Dynamics/Convair and Ryan Aeronautical Company in San Diego to review their efforts in this field. The results of this investigation are reported in Section 2.3.1. Since no boom was commercially available, the design decision was made to utilize a pneumatically deployed and mechanically retracted telescoping boom for this sampler prototype. In order to provide torsional stability and strength, the telescoping segments of this boom were required to be keyed together to prevent rotation about the axis of the boom. This is necessary so that sampler orientation with respect to the surface is predictably maintained. The design approach was to utilize available aluminum tubing that was made in a .058 inch wall thickness and stepped down in diameter in .125 inch increments. This resulted in a heavier boom than necessary, but was felt to be justified in terms of keeping fabrication costs and time to a minimum. A nine segment boom with 16 inch long segments was selected as being adequate to demonstrate the concept and satisfy the intent of the contract. This results in a boom which can extend to a total length of ten feet. The amount of overlap between tube segments was arbitrarily chosen to be twice the diameter of the larger telescoping segment. The adequacy of this overlap was demonstrated in some preliminary tests which will be discussed in the next section under fabrication. In order to allow the boom to operate without organic lubrication, teflon inserts were incorporated at the inner and outer surface at each end of the telescoping segments. Thus, the sliding interface is between a teflon and metal surface. Another advantage of using teflon inserts is that the breakaway and sliding friction is essentially the same. This characteristic should assist in providing a smooth deployment action to the boom.

Reliability and minimum weight were considered to be fundamental requirements in the design of the gear train and mechanism to control boom extension, elevation, and pneumatic valving. In order to maintain weight at a minimum, it was decided to use only one drive motor to provide power to the extension control tape drums and the elevation gear train. The block diagram for the basic gear train is shown in figure 4-19. In essence, this gear train represents a mechanical programmer which automatically provides the logic to perform the correct sequencing of functions. At the time that this gear train was designed, the criteria being observed was a requirement for a quick erection and deployment to the surface followed by a slow traverse over the surface. In retrospect, a closer examination of the block diagram indicates that a large degree of simplification could be achieved by extending and retracting at the same speed. This would eliminate the overrunning roller clutch on the extension gear train, although the latter might desirably be retained for safety purposes should the sampling head become lodged against an obstruction. Another advantage to this arrangement would be that a one to one gear ratio for valve actuation could be achieved by converting the extension gear train to a valve drive gear train. This would result in a faster opening action of the valve which is desirable from the standpoint of the gravity dump mode of delivering a soil sample. The valve currently closes in .072 seconds but requires 1.8 seconds to open. This is equivalent to a boom rotation from the vertical of about 11 degrees. This change would mean that the elevation reaction torque slip clutch would have to slip for about 10 revolutions if the existing retraction gear ratio and elevation gear ratio is retained.

The reaction torque slip clutch has been incorporated to accommodate the unknown or variable elevation angle to be traversed. The angle traversed from the extreme down position to the vertical is 140 degrees. Since it will probably be desirable to stow the sampler in a horizontal position and since the surface conditions at the terminal position of the sampler on the ground can vary, the slip clutch was incorporated to allow the elevation gear train to keep driving after the vertical stop position is reached. The vertical stop position will be reached before extension is complete when the boom starts from some position above the extreme down position.

Another option which was investigated in the design which could possibly simplify the boom is the use of another set of thin walled telescoping tubes concentric with the outer structure tubes to form the pneumatic transport tube. Two advantages of this arrangement would be the elimination of the valves at either end of the boom and a pneumatic transport tube with a less diverging cross section. This would allow the boom to be pressurized in the annular space between the two concentric sets of tubes and

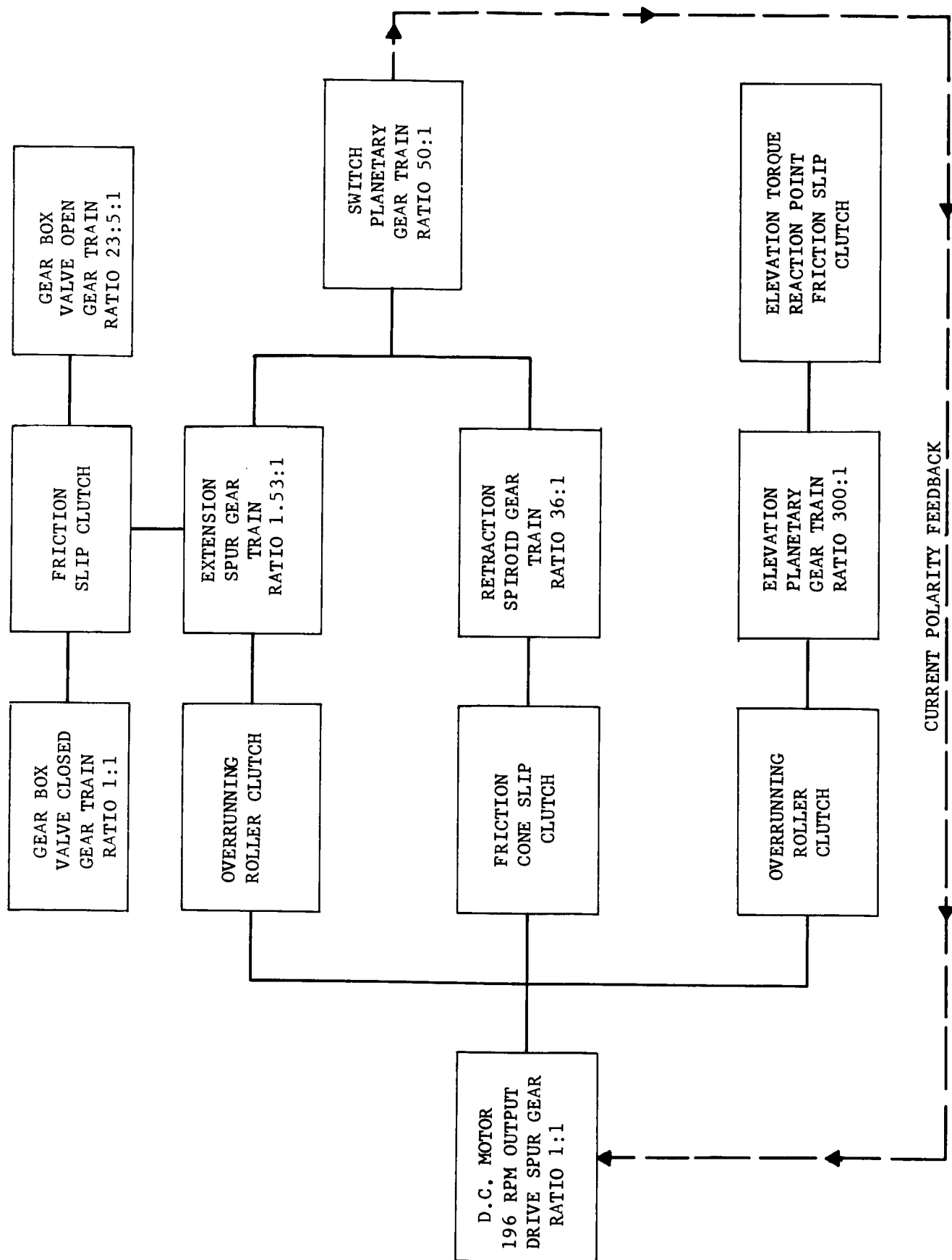


FIGURE 4-19 BLOCK DIAGRAM - TELESCOPING BOOM GEAR TRAIN

and would shield the structural sliding joints of the boom from soil particles. The inner set of tubes was not incorporated into the design because they would have to be extremely thin walled (in the order of .003 to .005 inches wall thickness). These were felt to present a fabrication time delay problem for the scheduled effort of the program.

In order to check the strength of the boom, a 1.5 pound sampler was assumed to be mounted on the end of the boom held in a horizontal position. The shear, moment, slope, and deflection diagrams were determined as shown in figure 4-20 for the boom extended in the horizontal position. The stress levels in the boom were very low. Shear stresses did not exceed 50 psi and bending stress did not exceed 2400 psi in the boom; however, it was noted that a reaction torque of 350 inch pounds is imposed on the base support structure and the elevation gear train. Since the pitch diameter of the ring in the planetary elevation gear train is 2.083 inches, the load reacted at the ring gear teeth is numerically equal to the applied moment. A diametrial pitch of 48 was selected as a reasonable compromise between gear size and gear tooth size. For this pitch, the allowable load per tooth is about 50 pounds. Since the final stage has two planetary gears, the total allowable torque that can be reacted is of the order of 100 inch pounds. This is inadequate under earth gravity conditions because it leaves about 250 inch pounds of torque to be reacted during deployment. This excess torque is reacted with a pneumatic snubber system consisting of two cylinder and piston assemblies connected to the boom through cranks attached to the supporting shaft. Initially, an attempt was made to use a spring load toggle system to provide this snubbing action; however, the size and force restrictions on the spring were not compatible. Figure 4-21 shows the variation of reaction torque as a function of the deployed angle from the vertical. It also shows the reaction torque with both types of snubbers applied. This torque is that which must be reacted by the elevation gear train. Figure 4-22 shows the variation of reaction torque as a function of boom length for various elevation angles. It is seen that the largest value of reaction torque occurs when the boom is extended to ten feet and is in a horizontal position. This reaction torque decreases as the boom is retracted so that at some point the snubber torque becomes greater than the moment caused by the boom and sampler. At this point the sampler would lift off the surface. This was another reason for selecting the pneumatic snubbing system since the pressure can be bled off after the sampler reaches the surface thereby reducing the torque or moment provided by the snubbers allowing the sampler to remain on the surface.

Since the overrunning clutches in the elevation drive gear trains transmit power in only one direction to lift the boom, there is no positive force or moment to start the boom down. In order to achieve this action a spring loaded plunger was incorporated into the vertical stop to produce a restoring moment. This plunger is also used to press the azimuth stop pin down

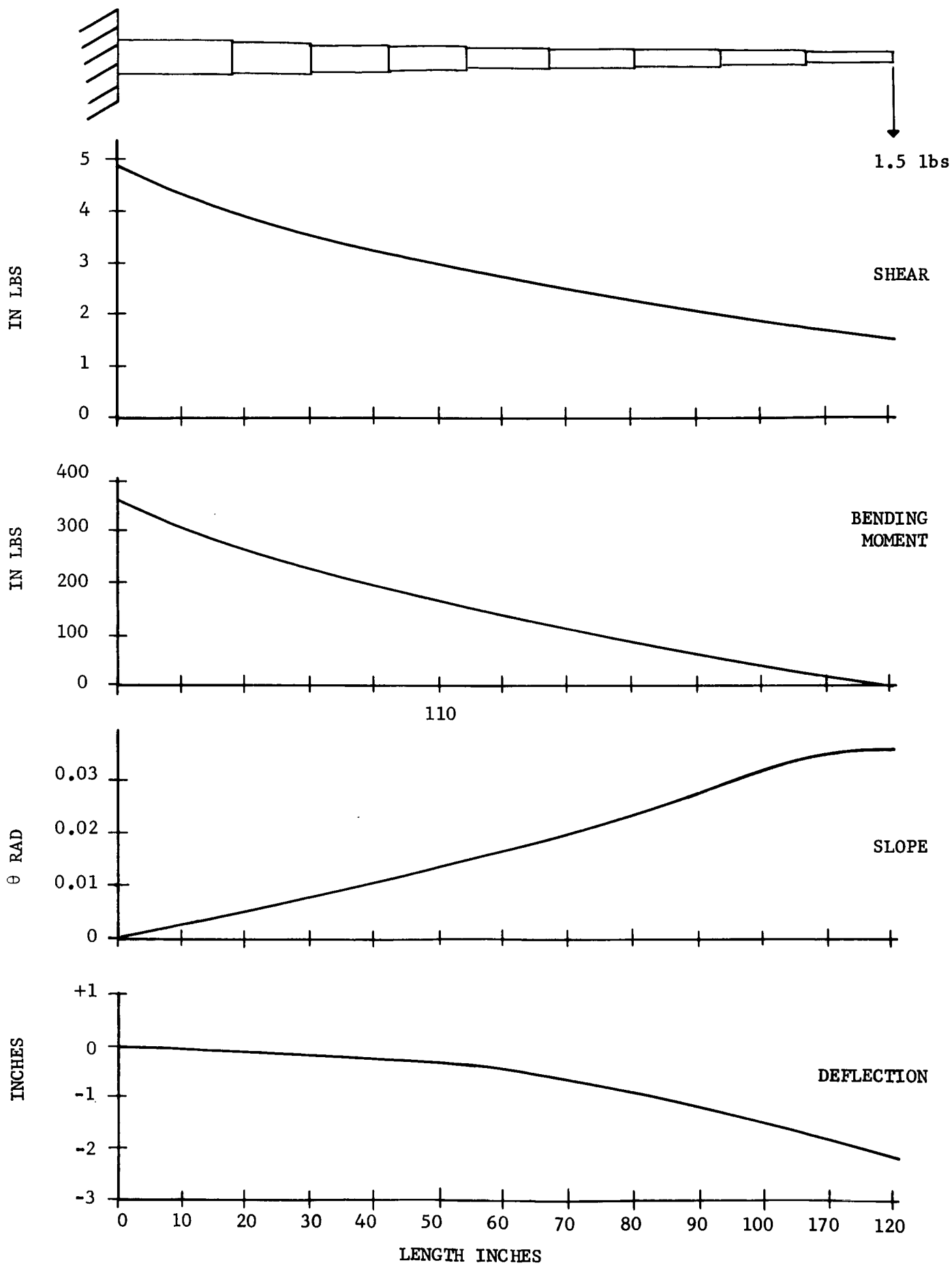


FIGURE 4-20 BENDING CHARACTERISTICS OF THE TELESCOPING BOOM

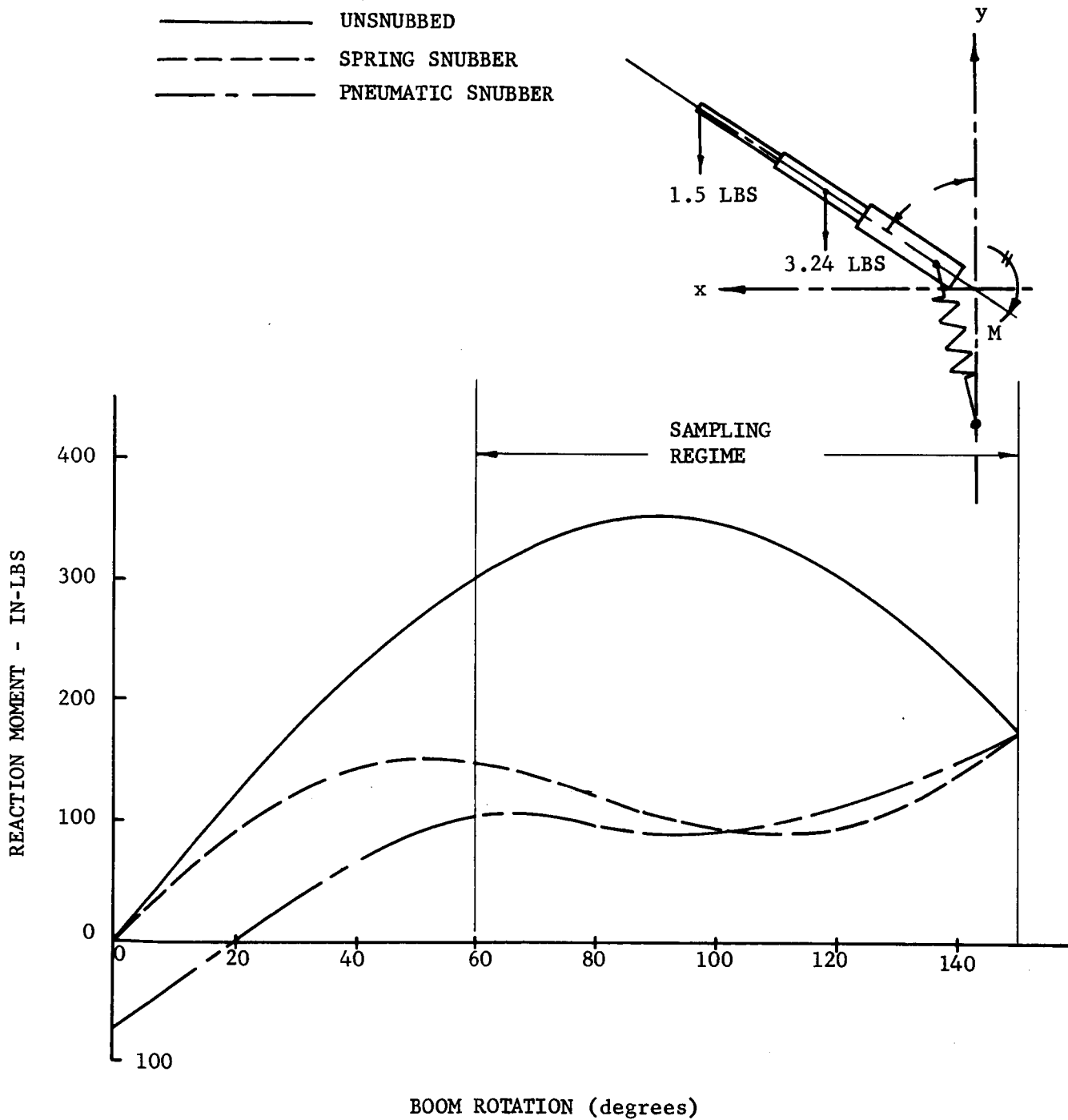


FIGURE 4-21 REACTION TORQUE AS A FUNCTION OF DEPLOYMENT ANGLE

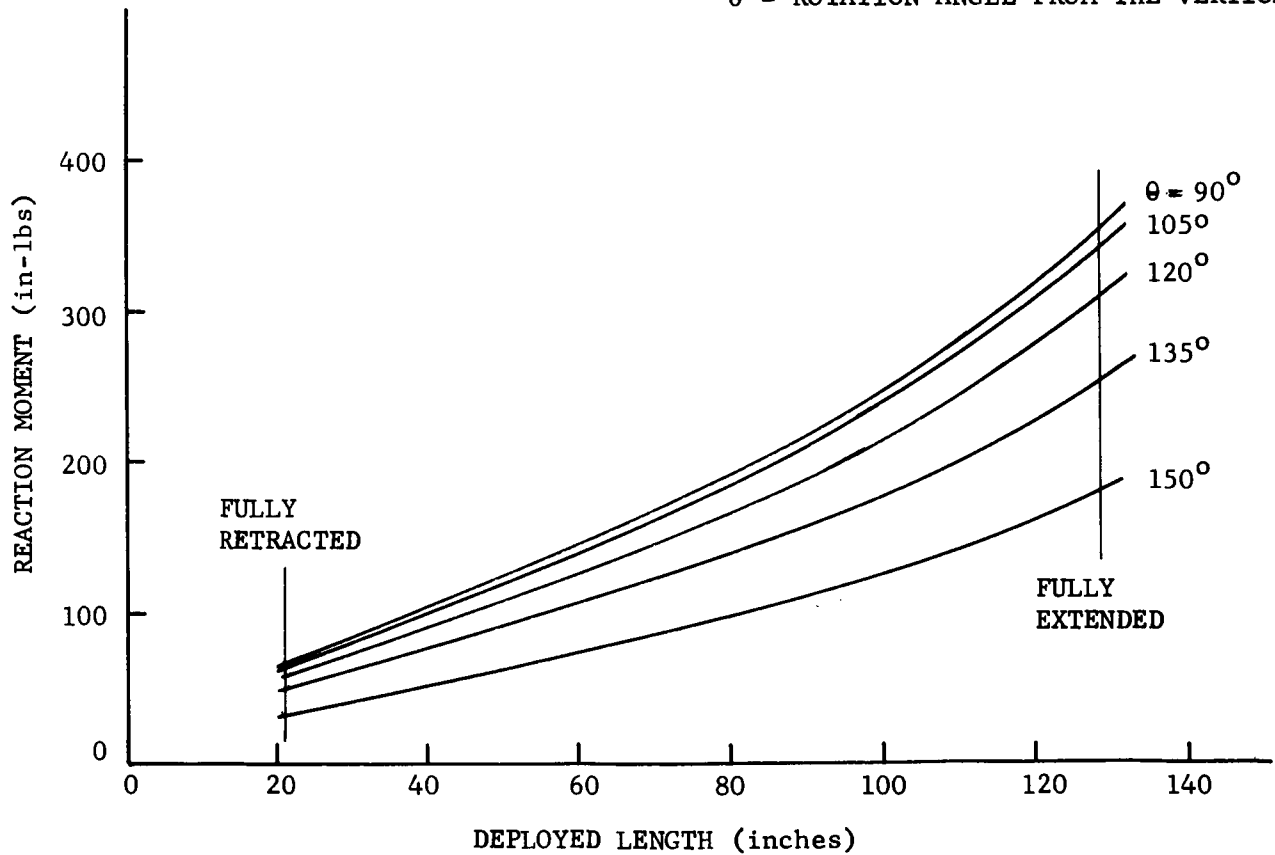
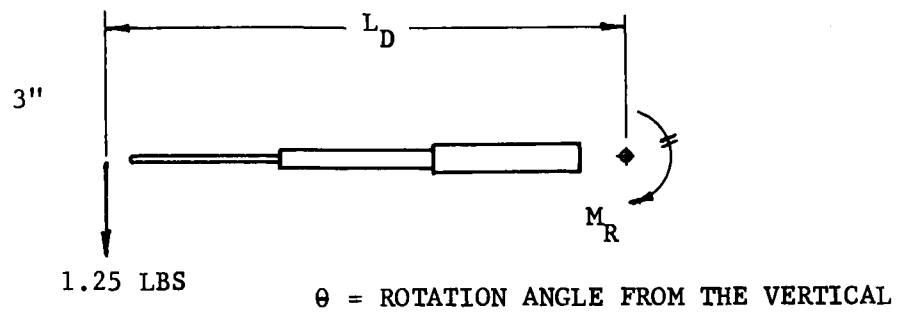


FIGURE 4-22 REACTION TORQUE AS A FUNCTION OF DEPLOYED LENGTH

in the support base allowing a spring drive to rotate the boom to a new azimuth setting on the subsequent sampling run. Also, in order to soften the action of the snubber near the vertical position, where they are not needed, and to assist in providing a restoring moment to start the boom down the snubber cylinder crank angles were located 65 degrees out of phase. At the vertical or erect position one cylinder has gone 35 degrees past dead center to produce a torque opposing the other cylinder which is 30 degrees before dead center. The combined effect of the vertical stop spring and snubber action is shown in figure 4-23 for the final prototype design. The shaded area represents the operating tolerance imposed by vertical misalignment of the sampler with respect to the surface. The angle θ_i is the angle between the boom axis when in the erected position and the local vertical. For this design the boom has a tolerance of ± 7 degrees. If the boom stops short of the vertical the tendency is to overload the gear train during deployment. If the boom goes past the vertical, the tendency is to trap it in this position since the moment produced by the boom weight opposes the restoring moment of the snubbers and spring. It is noted again that this is an earth gravity design condition. Under Mars gravity, the snubber system is not required and the vertical tolerance will be greater.

The remaining design effort on this sampler prototype is concerned with the rotating wire brush sampling head. Because of the high reaction torque or moment encountered with the boom, it was very desirable to keep the weight of this sampler to an absolute minimum since a weight on the end of the boom contributes most heavily to this moment. For this reason a 3 inch diameter brush was selected rather than the 4 inch diameter brush used in the breadboard development. The speed of the brush was increased to provide the same tangential velocity as was achieved in the breadboard tests. This results in a brush rotational speed of 437 rpm which produces a tangential velocity of 343 feet per minute. The final rotating wire brush design is shown in figure 4-24. The use of sample inlet ports over a 60 degree segment of arc rather than the 160 degrees used in the breadboard was determined by examining the sampler head geometry with respect to the surface for a 30 degree upslope, a 30 degree downslope, and a horizontal surface. It was discovered that in all cases the initial contact was nearly 10 degrees down with respect to the boom axis and changed as the boom is retracted to a maximum of 50 to 60 degrees below the horizontal. Since it is impossible to predict all types of surface conditions to be encountered, the location of sample inlet ports over this segment of arc was felt to represent an optimum compromise between simplicity and function. This also reduced the number of inlet ports which allowed the pneumatic flow area into the sampling head to be made compatible with the flow area in the small diameter of the transport tube. This serves to maintain a higher velocity flow through the head where the soil is acquired. This area was not optimal in the breadboard model; however, an order of magnitude increase in sample size collected on the hardpan model was noted when pneumatic transport was used.

BOOM AXIS

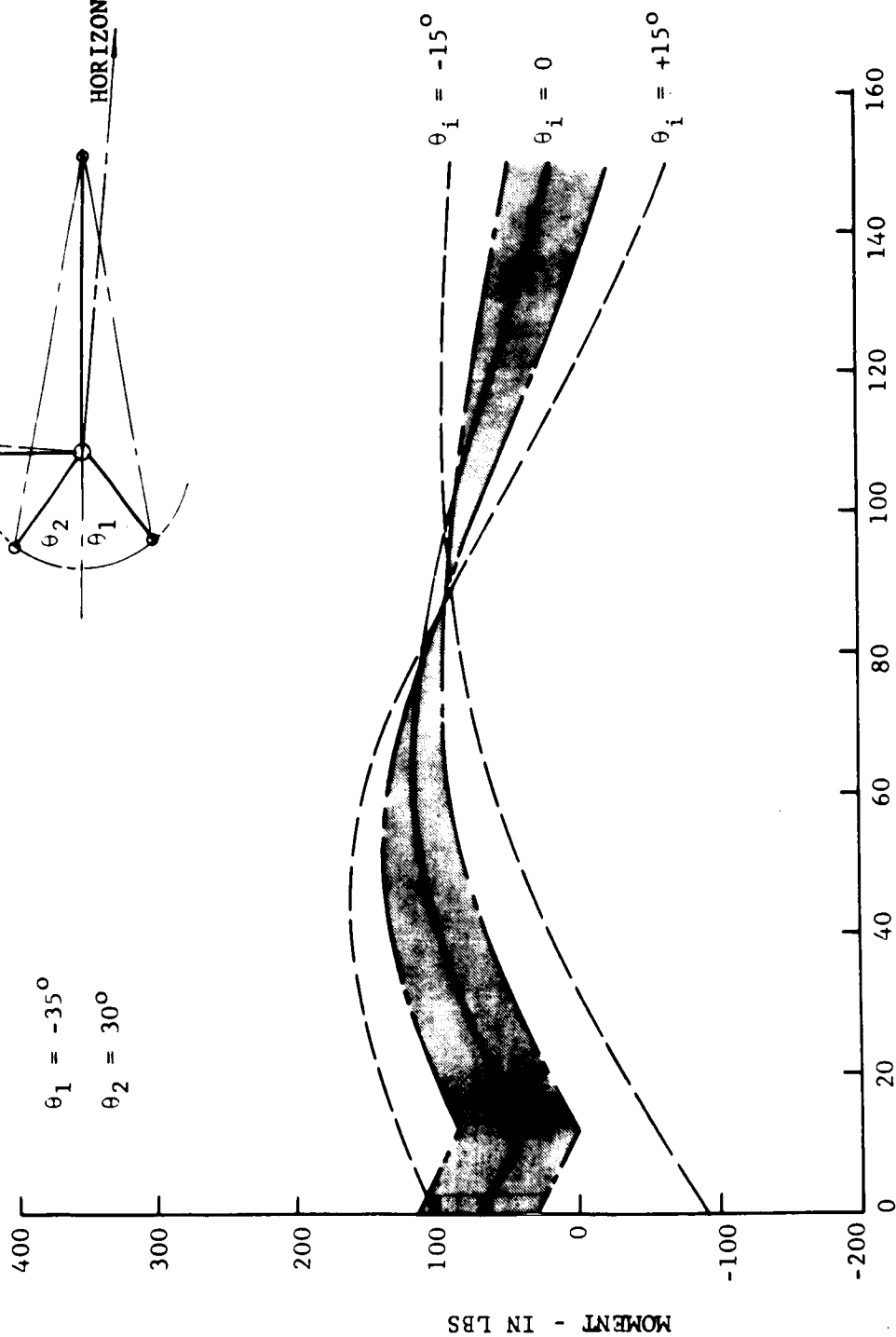
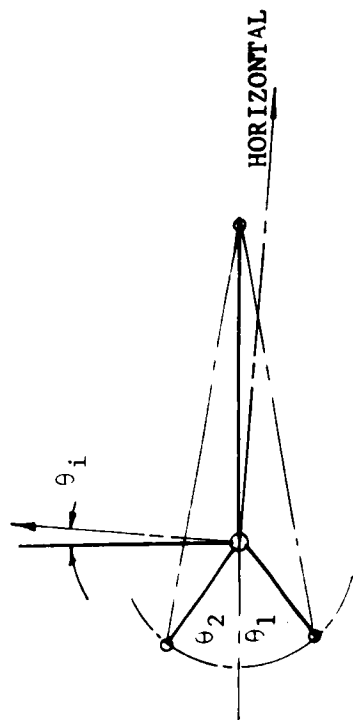


FIGURE 4-23 REACTION TORQUE LOAD IMPOSED ON GEAR TRAIN

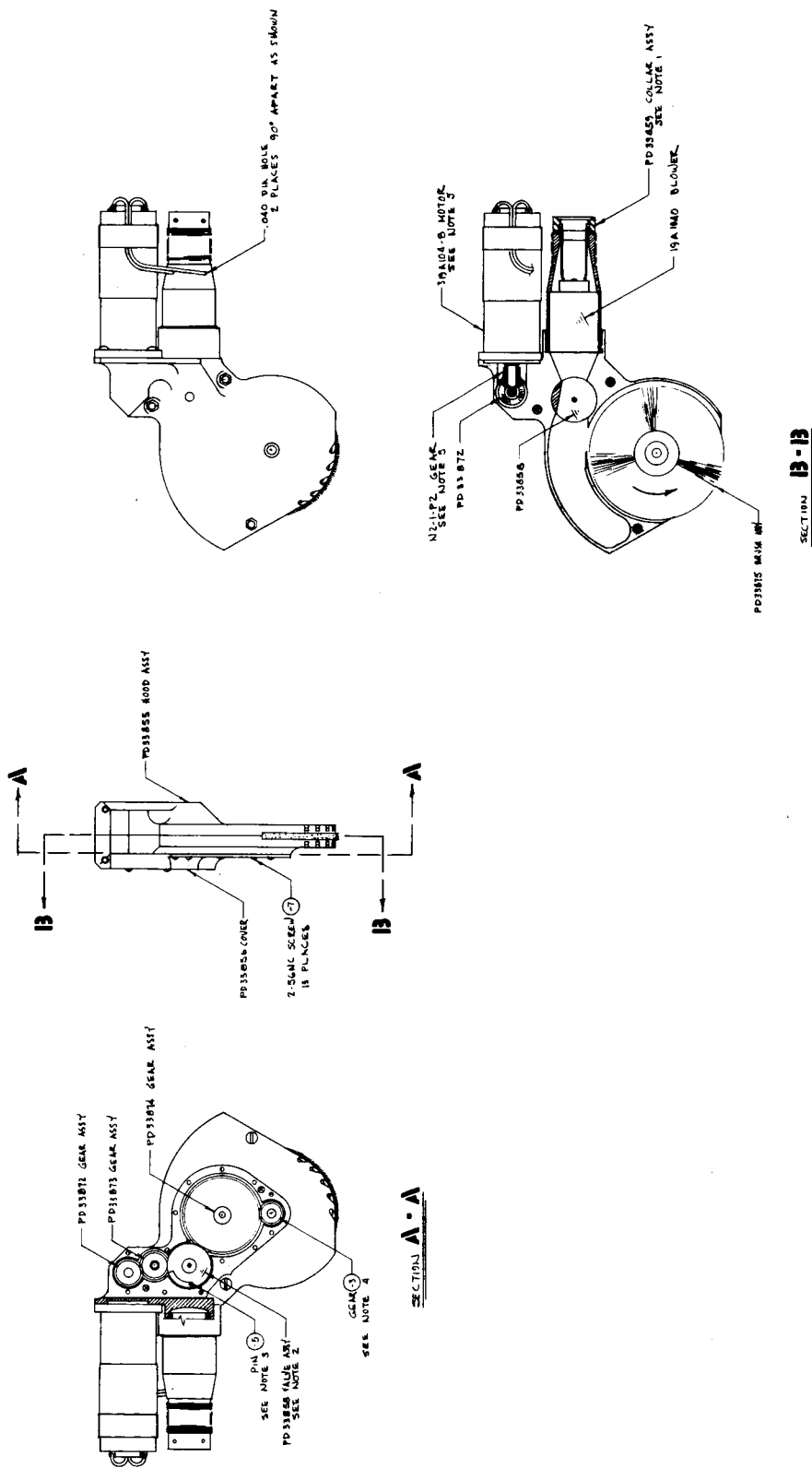


FIGURE 4-24 ROTATING WIRE BRUSH PROTOTYPE ASSEMBLY

As was noted in Section 3.3, considerable power was dissipated due to the internal configuration of the hood and the wire brush. The interim configuration was revised to provide a minimum of contact between the bristle ends and the side of the housing. Also, the soil entry ports were machined so that a continuous rub strip is left on the inner edge, thereby eliminating the condition where the abrading bristles fall into these ports and must be forced out again during rotation. Because the breadboard testing with pneumatic collection showed an order of magnitude increase in the amount of soil collected on severe models such as hardpan, this sampler was designed to provide both mechanical collection and pneumatic collection of the soil. This will provide a redundancy in the sampling method which should result in a higher probability of collecting a sample. While the single stage vaneaxial blowers will not be adequate at 5 millibars, the design lends itself to very easily incorporating a multistage blower. The valve assembly shown in figure 4-24 is used to seal the end of the telescoping boom during the pneumatic extension of the boom. This valve is actuated through a slip clutch running off of one of the idler gears in the brush drive train. Reversing the brush rotation will close this valve. A gear train was used to drive the brush because it was the most compact in terms of depth of housing required and being adaptable to valve actuation. In the gravity dump mode, the soil falls on top of the valve during boom erection. When the boom reaches the vertical position, the motors are reversed, opening the valve which allows the soil sample to fall down the tube through the blower. Complete valve opening is achieved in about .07 seconds.

The final aspect of the rotating wire brush/telescoping boom prototype was the delivery of power to the drive motor and blower mounted in the sampling head. It was natural to think of using the extension control tapes as electrical power leads to these motors. This introduced the problem of providing power to the tapes through a rotating member. A cross sectional view of the upper and lower extension control tape assemblies are shown in figure 4-25. In both cases, an insulated conductor mounted concentric with the shaft protrudes through the side of the gear box housing where contact is made with a beryllium copper brush. At the tape spool another conducting pin is pressed into the insulated concentric conductor. In the case of the lower tape assembly, this conducting pin is soldered to the tape spool to which the tape is pinned. In the upper tape assembly, the friction clutch must slip during extension of the tape requiring another sliding brush contact. To ensure contact at all times, the clutch and tape drum are keyed to the shaft so that they may float axially along the shaft. The clutch and tape spool assembly are preloaded against the sliding brush or pin contact with a small belville spring. The tapes are coated to provide insulation to prevent them from shorting against each other or the boom walls. The ends of the tapes are pinned to an insulating collar mounted between the end of the boom and the sampling head. This collar is shown in figure 4-24

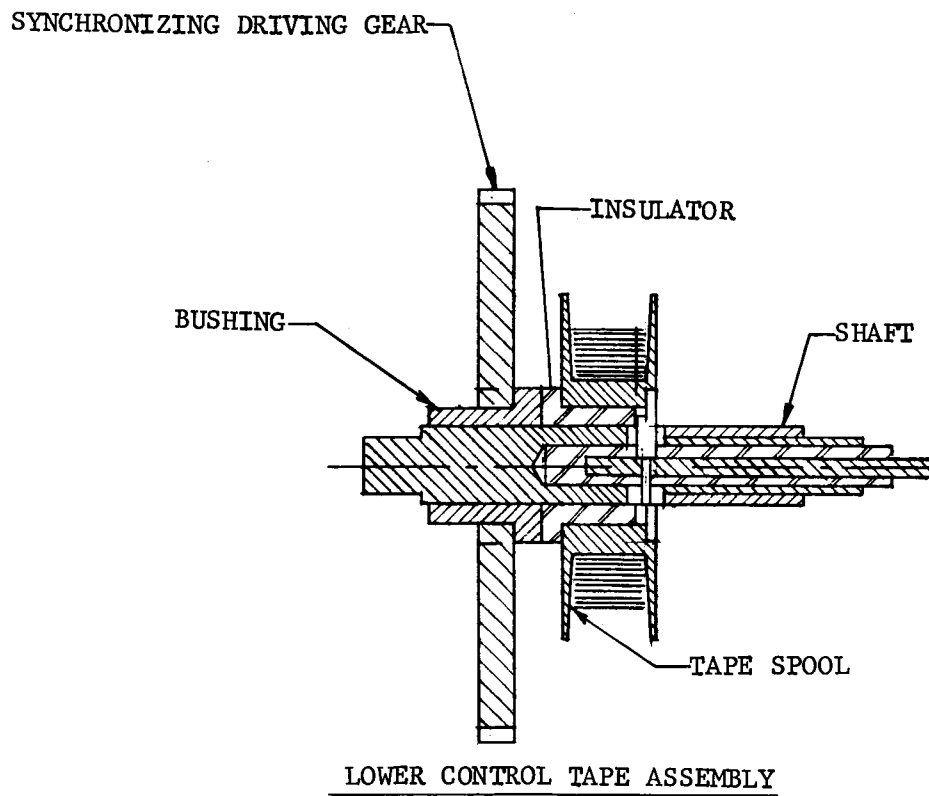
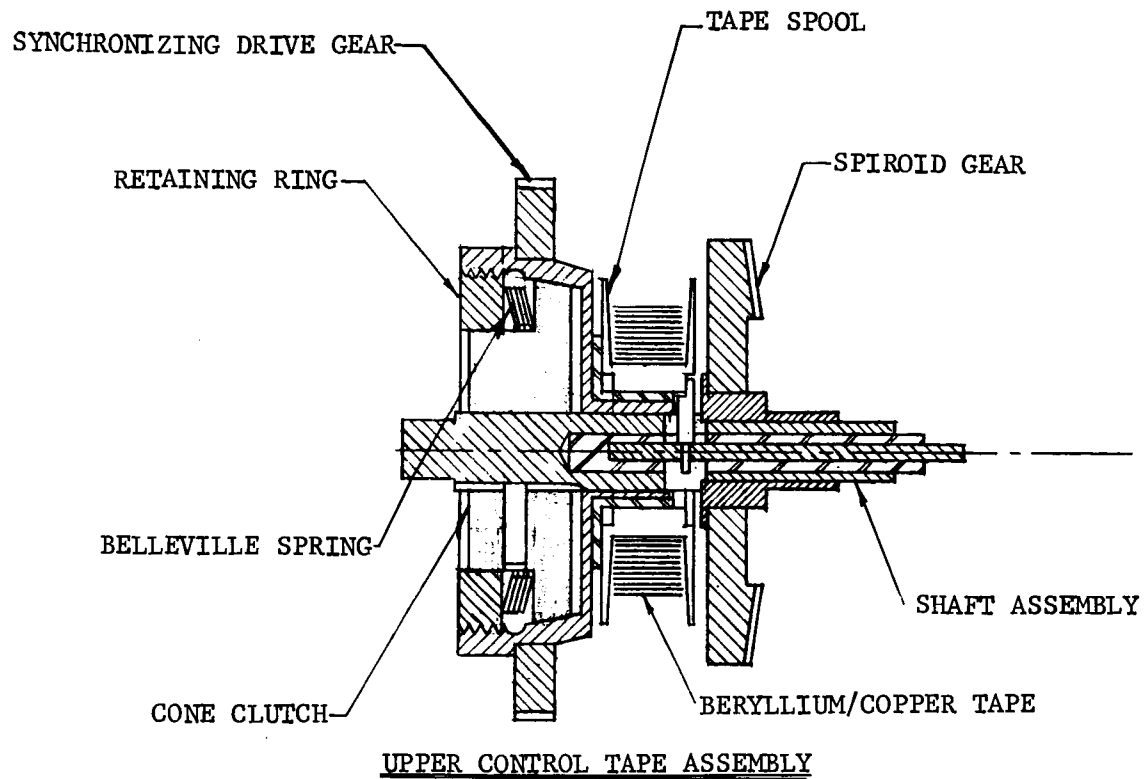


FIGURE 4-25 CROSS-SECTIONAL CONFIGURATION OF EXTENSION CONTROL TAPE ASSEMBLIES

of the prototype sampler assembly. Small brass tubes are mounted in this collar to which the power leads from the blower and the drive motor are soldered. When the tapes are pinned in the collar electrical continuity to the sampling head is achieved.

The total weight of the rotating wire brush/telescoping boom sampler prototype as developed in this design is 11.5 pounds. A weight breakdown by components and subassemblies is given in Table X. It is estimated that this sampler can be constructed for about half this weight in a next phase design by utilizing fiberglass or magnesium in the boom and using aluminum gears in place of brass and steel.

4.2.2 FABRICATION

The initial task was to verify the telescoping boom design using readily available aluminum tubing. This was done by first fabricating three typical boom segments rather than a complete set of nine.

The segments of the boom were fabricated by bonding in teflon inserts at the ends of the aluminum tube segments. These are machined after bonding to fit the adjacent tube segments. No difficulty in fabrication of these segments was encountered. The fit between each segment was deliberately left fairly tight to minimize cocking of the segments when extended. This three segment portion of the boom was mounted in a fixture and pneumatically deployed with controlled pressures. A minimum pressure of 3 psi was required to achieve full deployment; however, 4 psi produces more consistent deployment.

When pressure was applied, the tube extended slowly and relatively smoothly although the rate varied slightly as the tightness of fit varied. At no time did the rate increase abruptly or continuously. This can be attributed to the friction characteristics of teflon which has a breakaway friction coefficient approximately equal to the sliding friction coefficient. The pressure was varied to determine the effect on extension rate. These results are shown in figures 4-26 and 4-27.

Figure 4-26 shows the effect of repetitive cycling on the deployment rate at a constant pressure. It is seen that the rate increases as deployment time decreases asymptotically towards a minimum value fairly rapidly. This is probably due to the burnishing action accompanying each extension. Figure 4-27 shows the effect of varying the deployment pressure. Again, it is seen that the deployment time decreases rapidly with pressure and also asymptotically approaches a minimum deployment time. This trend indicates that an increase in deployment pressure above some value is not necessary and will only increase the energy of impact when full extension is achieved.

TABLE 4-X

ROTATING WIRE BRUSH/TELESCOPING BOOM SAMPLER
PROTOTYPE WEIGHT BREAKDOWN

Component	Component Weight	Assembly Weight
Wire Brush Prototype	.71	
Motor	.72	
Sampler Assy Total		1.43
Telescoping Tubes	3.25	
Gearbox Assy	1.87	
Motor	.72	
Support Assy	3.32	
Snubber Cylinders	.43	
Elevation Clutch	.48	
Boom Assy Total		10.07
Prototype Assy Total		11.50

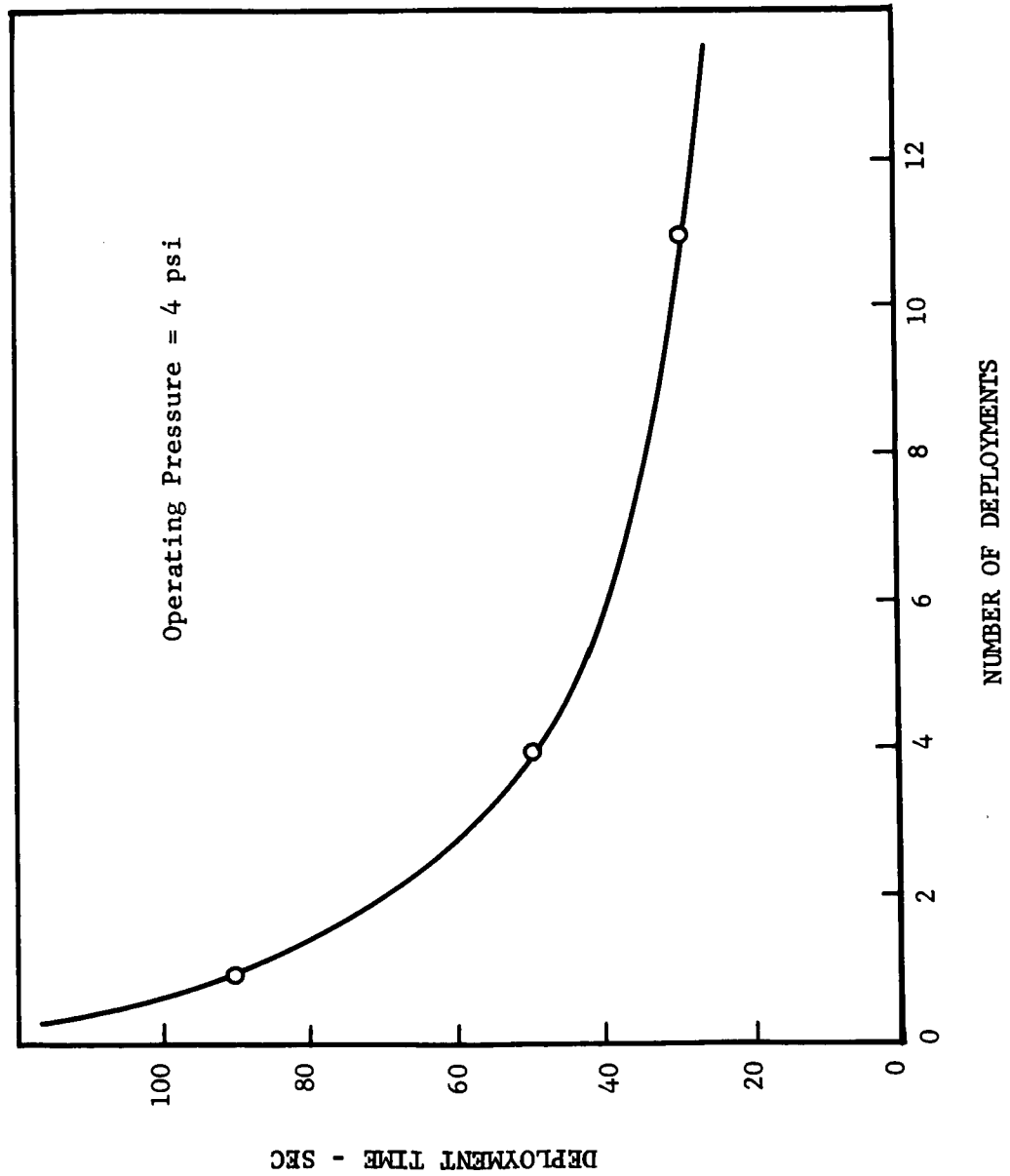


FIGURE 4-26 BOOM DEPLOYMENT TIME WITH REPETITIVE OPERATIONS

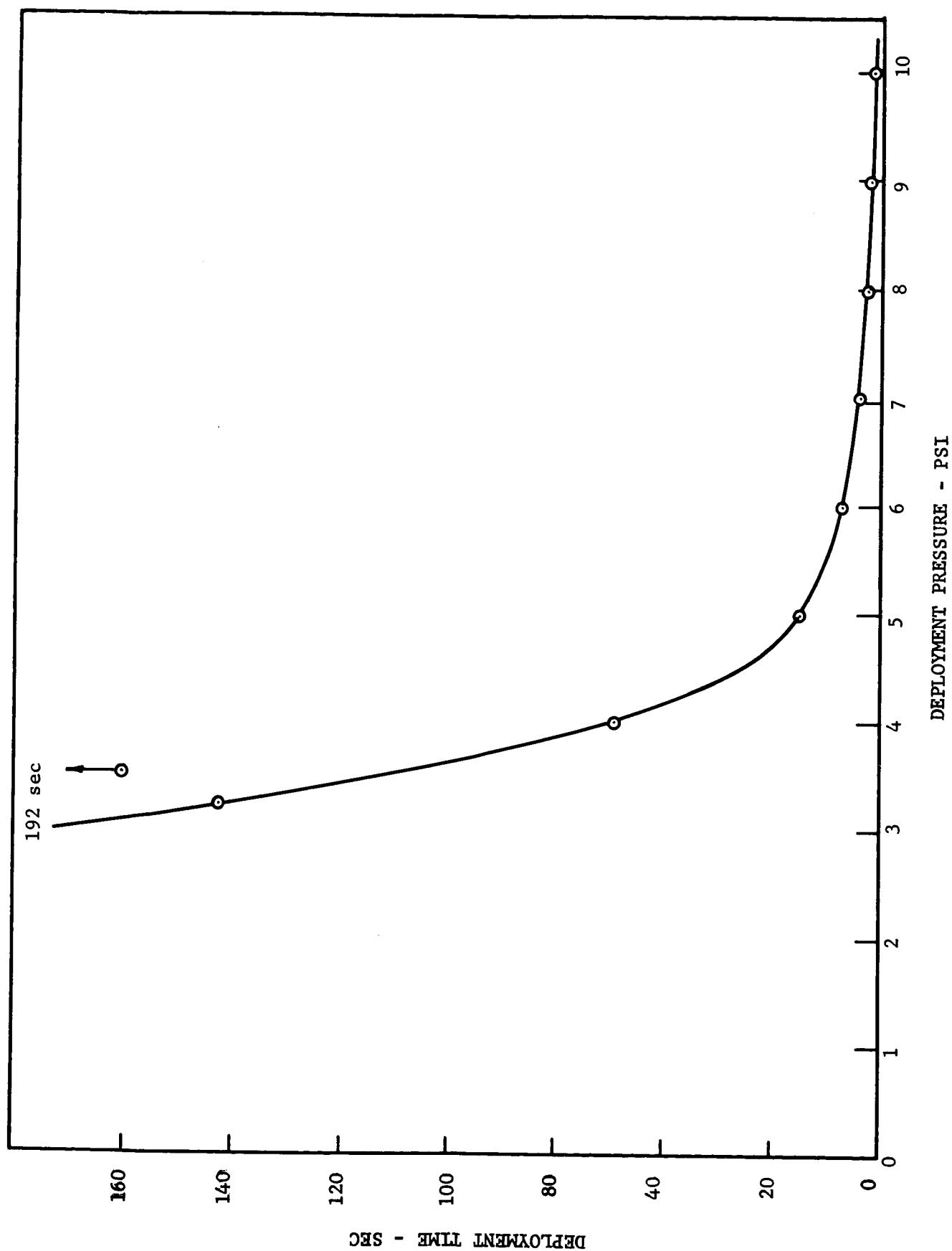


FIGURE 4-27 BOOM DEPLOYMENT TIME WITH INCREASING PRESSURE

To check the effect of cold flow on the teflon inserts, the three segment boom was loaded to reproduce the actual moments at the joints between the segments for the complete boom. The load, shear, and moment diagram for this test are shown in figure 4-28. The boom was loaded and the initial deflections noted, as well as those occurring after 24 hours under load.

The data is summarised in Table 4-XI

TABLE 4-XI
TELESCOPING BOOM JOINT COLD FLOW TEST

<u>Deflected Height Of Boom Tip-Inches</u>	<u>Test Time (Hrs.)</u>	<u>Condition</u>
28.633	0	Boom Weight only
28.025	0	Boom loaded
28.610	24	Boom loaded

It is seen that a permanent set of 0.023 inches occurred in the two joints. This set should be fairly uniformly divided between the two joints since the more heavily loaded joint also has slightly more overlap. The overlap used is twice the diameter of the larger tube segment. Deployment checks were run before and after the cold flow test and no effect on deployment operation was detected.

The three segments of the telescoping boom tested were returned to the shop for additional fabrication. A second attempt to assemble these segments resulted in some galling between the aluminum tubes. It is felt that this was caused by surface chips or residual contaminants from the machining. To prevent galling in actual operation, the clearance between the tubes was increased by machining the outer surface. After machining and polishing, the tubes were hard anodized to provide a surface less susceptible to abrasion. Three sets of tubes were fabricated to allow for internal configuration variations and to provide backup in the events of damage. After hard anodizing the tubes a graphite impingement type coating was applied to provide a dry lubricating film at the sliding surfaces. No galling or siezing was subsequently encountered. In order to hold weight to a minimum, magnesium was used in such places as the gearbox housing and the sampling head shroud. This material machines nicely and no problems were encountered due to its use. It may, however, result in fairly rapid wear wherever the rotating wire brush makes contact with the hood. This would be more of a problem in an extended testing program rather than use during an actual mission.

The beryllium copper extension control tapes were fabricated initially from .002 inch thick material and heat treated in a coiled configuration. This proved to be troublesome since they would twist into a helix if the tension

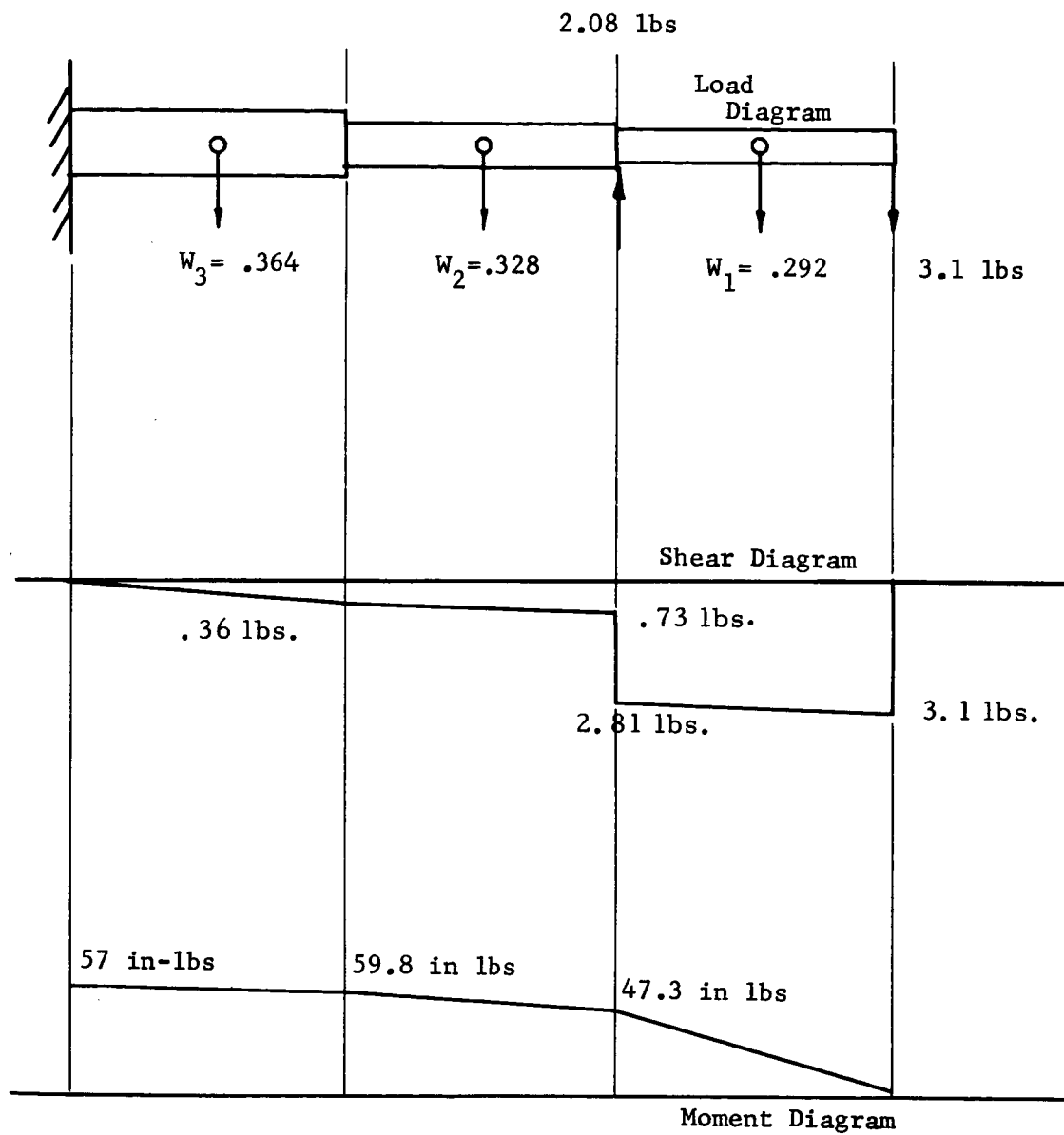


FIGURE 4-28 BOOM LOADING FOR COLD FLOW TEST

applied during extension were relaxed. This resulted in a snarled tape. It was also noted during handling that the end loops were easily broken off due to the brittleness of the material. The end loops were formed as shown in figure 4-29.

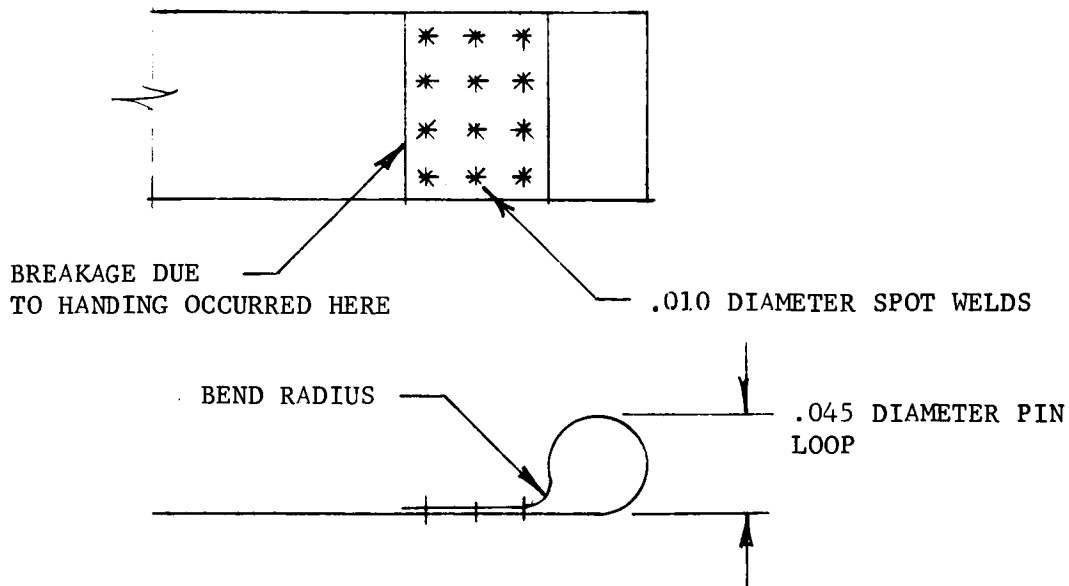


FIGURE 4-29

TAPE END LOOP CONFIGURATION

Breakage due to handling invariably occurred at the first row of spot welds. Several test tapes were made and tested. Failure occurred at less than 30 pounds force which was about half the theoretical yield point of the basic material. Two types of failures were observed. One was a fracture in the bend radius adjacent to the last row of spotwelds. The other failure mechanism was one in which the spotwelds were pulled out of the tape. In either case failure occurred at about the same value. To check the basic tape material, one tape was mounted in the testing machine without using the end loops and pins. The yield point and ultimate strength agreed very well with the theoretically calculated values. A new set of test tapes were fabricated with a larger bend quarter hard condition. Failures for the heat treated tapes occurred in the bend radius and for the as received tapes in the spotwelds. In both cases the failure load was still equal to or less than 30 pounds. Since this did not provide an adequate margin for the design, .003 inch thick material was procured

and tested. These tapes failed in a range from 35 to 45 pounds which was considered to be adequate. As before failure occurred in the spotwelds at the loops.

To provide electrical insulation, several tapes were coated with a variety of plastic coatings. The prime intent was to obtain an insulating coating with a minimum of build up on the thickness of the tape. In all cases insulation of the flat faces was achieved; however, the edges remained open. The nitrile phenolic coating originally tried required a heat cure of 30 minutes at 250°F. This was sufficient to induce enough heat treat to cause the tape to curl. This was unsatisfactory due to the snarling problems it caused. It might be worthwhile to mention at this point that this can occur during the dry heat sterilization cycle. The substitution of steel tapes for the beryllium copper tapes would avoid this problem.

A room temperature cure coating of polyurethane was finally selected. This was a commercial formulation known as Chemglaze Z052 clear gloss top coat diluted 25 percent by weight with toluene. The coating was applied by hand brushing and allowed to cure overnight. Although some runs occurred on the surface, no problems were experienced later in passing the tape smoothly through its teflon fairleads. In a similar manner, the open edges did not cause any problems with shorting. Under most conditions the tapes are under tension and do not touch the walls of the telescoping boom or the gearbox housing.

The gear train mechanism did not present any problems that are not normally present in the form of fits and tolerance accumulations except for the conical slip clutch which is used to allow rapid extension of the boom. This clutch is mounted in the upper tape assembly that is shown in figure 4-25 in Section 4.2.1. The conical clutch was fabricated of brass and runs in a stainless steel housing. Difficulty was experienced in setting this clutch up to slip at the correct value. Since the static coefficient of friction is higher than the running value, the breakaway torque is higher than the running torque. Thus, if the clutch is adjusted to give a reasonable breakaway value, the running torque is too low and slippage occurs. If the running torque is adjusted to the desired value, the breakaway torque tends to climb with usage until siezing occurs. It appeared that this was a place where teflon could be used to advantage. In order to prevent relaxation of the clutch torque pure teflon cannot be used since it has a tendency to cold flow under load. A clutch facing was fabricated of a ceramic loaded teflon which was bonded to the clutch face. The characteristics of this clutch are much smoother with a running torque that tends to increase as the slip velocity increases. No apparent attempt to sieze was noted when the clutch was burnished in on the bench and adjusted to the desired torque of 15 inch pounds.

In the first attempts to pressurize the telescoping boom severe leakage was encountered. The possible sources of leaks are at the sliding joints between the boom segments, the valves at either end, the teflon fairleads

where the extension control tapes leave the gearbox housing and enter the boom, and through the boom to gearbox joint. The leakage at the valves has to be tolerated since these have been machined to as close a fit as is practical for the actuation loads available. The boom to gearbox joint was sealed with a non hardening mastic.

The fabrication of the rotating wire brush sampler was fairly straightforward. The hood was machined out of two pieces of magnesium which are doweled and bolted together. The only problem encountered was the same as was noted during the breadboard envelopment testing in that considerable power was lost internally. The wire brush used is a commercially available brush which had a fairly full set of bristles. By trimming away some of the side bristles the power loss was reduced to the point where it was acceptable. In use the bristles will eventually flare with the result that power losses will gradually increase as was noted in the breadboard tests. It is probable that a special brush with fewer bristles of heavier wire should be fabricated to minimize internal power losses.

4.2.3 PERFORMANCE EVALUATION

Due to delays encountered in the rotating wire brush breadboard hardware and testing, sufficient time in this program did not remain to allow testing according to the proposed matrix shown in figure 4-30. The primary effort expended in testing this prototype has been to operate the mechanism in various modes in order to correct deficiencies that have become apparent.

The primary problem area which remains to be solved is that of pneumatically deploying the telescoping boom. In functional tests it was discovered that leakage at the valves and the sliding joints of the tube segments was excessive. Air pressure supplied at 100 psi through an eighth inch diameter tube did not produce a flow sufficient to cause a detectable pressure rise within the boom. Evidently the teflon inserts are not thick enough or resilient enough to provide an effective seal along the length of the tube. Since there are nine tube segments, this results in eight sliding joints which will require that leakage at each joint be minimized. The addition of an elastoemeric seal mounted in a ring which is pressed into the end of each tube segment is shown in figure 4.31. This would be a simple modification which would probably eliminate leakage at the sliding joints.

The gearbox has been run several times and functions as desired. It is noted that the friction cone clutch is not entirely predictable. In the process of disassembly and reassembly to trace out malfunctions and de-bugging the system, the performance of the clutch will vary. In some cases it will

SOIL MODEL		3	3	3	3	3	3	3	1	2	4	3
ENVIRONMENT		A	A	A	A	A	A	A	A	A	A	V
TEST CONFIGURATION		1	2	3	4	5	6	7	8	9		
VARIABLES												
SAMPLING HEAD ATTACK ANGLE	15°											
	30°											
	45°											
SOIL TRANSPORT MODE												
GRAVITY PNEUMATIC COMBINATION	GRAVITY											
	PNEUMATIC											
	COMBINATION											
SOIL MODEL 1												
2												
3												
4												
ENVIRONMENT A												
V												
ATMOSPHERE												
VACUUM												

FIGURE 4-30 ROTATING WIRE BRUSH AND TELESCOPING BOOM TEST MATRIX

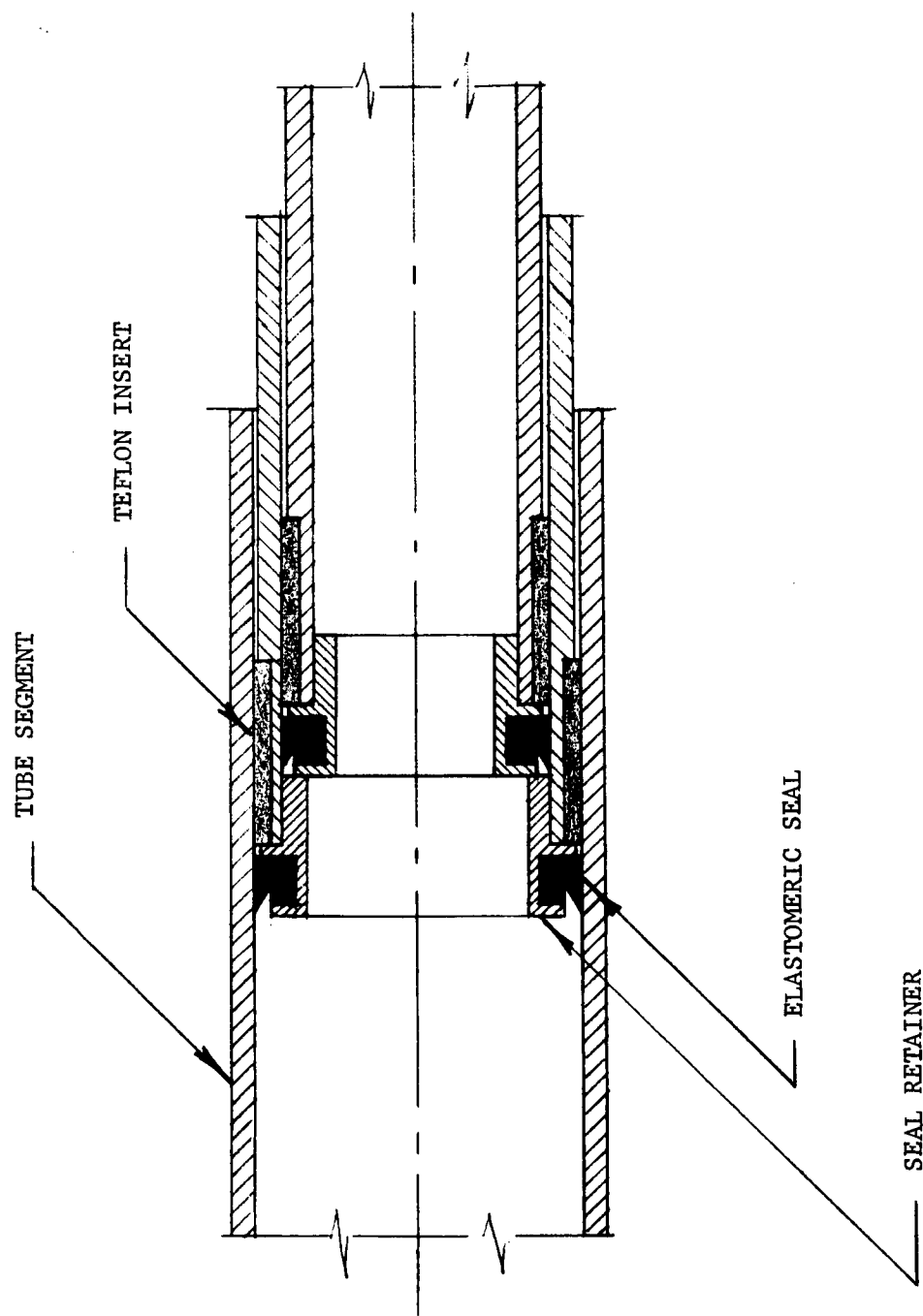


FIGURE 4-31

FIGURE 4-31 POSSIBLE PNEUMATIC SEAL FOR THE TELESCOPING BOOM

deploy the tapes smoothly but will appear to slip on retraction. In other cases it tends to overload the motor during extension because the torque required to slip becomes excessive. In the interest of reliable operation it would appear that the high speed extension should be eliminated. In this case the spiroid gear would drive the tape spool in both directions eliminating the need for the slip clutch. This is very simply accomplished by removing the high speed extension drive gear from the gear train. The elevation gear train operates as predicted and will present no problems in operation.

SECTION 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSIONS

The effort in the development of the two types of engineering prototype samplers is summarized in the conclusions in the following paragraphs. This development effort was broad in scope and encompassed a wide variety of design problems associated with automatically operated remote soil samplers. The solution of many of these problems is felt to have contributed to the state-of-the-art and knowledge concerning remote planetary soil sampling systems for biologically oriented missions.

5.1.1 VERTICALLY DEPLOYED CONICAL ABRASIVE SIEVE PROTOTYPE

This sampler system is suitable for development into an early planetary mission flight hardware. The engineering prototype evaluation has demonstrated the feasibility and to some extent the simplicity and reliability of operation. The following specific conclusions are listed.

- (1) This sampler can be developed into a flight prototype weighing less than 5 pounds and requiring less than 25 watts of power.
- (2) Sampling can be accomplished in a large variety of soil types under a variety of surface conditions.
- (3) The basic drive and feed mechanism can be easily adapted for use with other cutting heads or augers to provide a versatile assortment of samplers for specific uses.
- (4) Soil sample sizes up to 25 grams per run can be collected. With modifications even larger samples are possible.

- (5) This sampler system is capable of abrading the surface sufficiently aggressively to collect samples from hardpan and very soft or decomposed rock models.
- (6) The feed rate of the drive mechanism is self limiting and automatically adjusts the rate to be compatible with the progress of the cutting head while maintaining a positive force between the cutter head and the surface being sampled.

5.1.2 HORIZONTALLY DEPLOYED ROTATING WIRE BRUSH/TELESCOPING BOOM PROTOTYPE

This soil sampling prototype system is a more versatile and sophisticated sampling system which can be suitably developed for early planetary missions. While the limited performance evaluation testing has not completely verified the concept of the pneumatically deployed telescoping boom, the extensive breadboard testing accomplished with the wire brush sampling head indicates that this sampler will be a versatile and efficient sample collector on a large variety of soils and surface conditions. The basic concept of remotely deploying a sampler system and sampling over an extensive area of the surface is inherently more complex than single point sampling thereby involving a greater development effort. The specific conclusions for this sampling system are presented as follows:

- (1) The rotating wire brush is very efficient at abrading nearly all types of surfaces to be encountered in soil sampling, including soft rock.
- (2) A combination of pneumatic transport and soil collection with the simultaneous mechanical collection of larger soil particles is highly desirable.
- (3) Multistage vaneaxial blowers will be required to effect soil transport by pneumatic means at reduced pressures in the range of 5 to 10 millibars.
- (4) The feasibility of the boom retraction and elevation mechanism has been established. Some additional changes and testing are required to establish the practicability and reliability of pneumatic boom deployment.
- (5) This sampler system provides a compact mechanism which can be deployed to ranges up to 10 feet away from the payload and sample continuously over an extensive surface.
- (6) This sampler prototype can be developed to weigh less than 10 pounds utilizing approximately 25 to 50 watts of electrical power.

5.1.3 CONCEPTUAL DESIGN EFFORT

This effort indicates that there are many methods of attacking the problem of soil sampling at remote locations. It also has shown that many gaps still exist in the development of adequate soil sampling hardware and

concepts depending on the mission constraints which are applied. Until this effort was initiated, the more sophisticated soil samplers were oriented towards collecting samples for geological analysis. Early efforts at collecting soil samples for biological analysis were intimately tied to specific experimental mechanizations with severe constraints on simplicity, and light weight, and small volume. Most of these sampling devices are inadequate to provide the necessary samples for more elaborate experiments requiring larger quantities of sample and are in most cases one-shot devices.

5.2 RECOMMENDATIONS

With the completion of this program, the necessary first step in developing a system prototype has been taken. This has consisted of designing and developing the engineering prototypes which is required before any systematic test program can be initiated. The primary emphasis in this phase of the development of potential flight hardware has necessarily been confined to making the system function. Many questions regarding the quantitative performance, operational characteristics, and mission suitability remain to be answered. It is therefore the recommendation of the Space and Reentry Systems Division of Philco-Ford at Newport Beach that the valuable data collected and effort thus far expended be enhanced and augmented by continued evaluation of these engineering prototype models including testing in the field. Also, since these soil samplers are intended to sample for biological missions, it is necessary to initiate some effort in determining the interaction of the soil sampler mechanism with the biological content of the soil to detect any biases or biological degradation that may occur.

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APPENDIX A

BIBLIOGRAPHY

The Aeronutronic Technical Library Staff was requested to conduct a literature search for review by Philco-Ford design engineers. A selected bibliography culled from the title search is here presented. The literature search sought to cover the following subjects from which preliminary scientific and engineering data could be abstracted for utilization as basic principles for the generation of sample handling system concepts:

(1) Soil Characteristics

- Soil properties
- Soil mechanics
- Soil simulation
- Particulate soils
- Cohesive soils
- Jet impingement on soils
- Soil interaction to impacts

(2) Engineering Design of Space Instruments

- Sample collecting, transporting and processing devices
- Soil mechanics instruments
- Drills
- Analytical instruments

(3) Structures for Space Use

- Engineering design
- Booms
- Pneumatic tubes
- Capsule geometries

(4) Organisms and their Detection

- Character of microorganisms
- Detection techniques and instruments

Despite this broad coverage, it is immediately apparent that the quantity of directly pertinent material is rather sparse.

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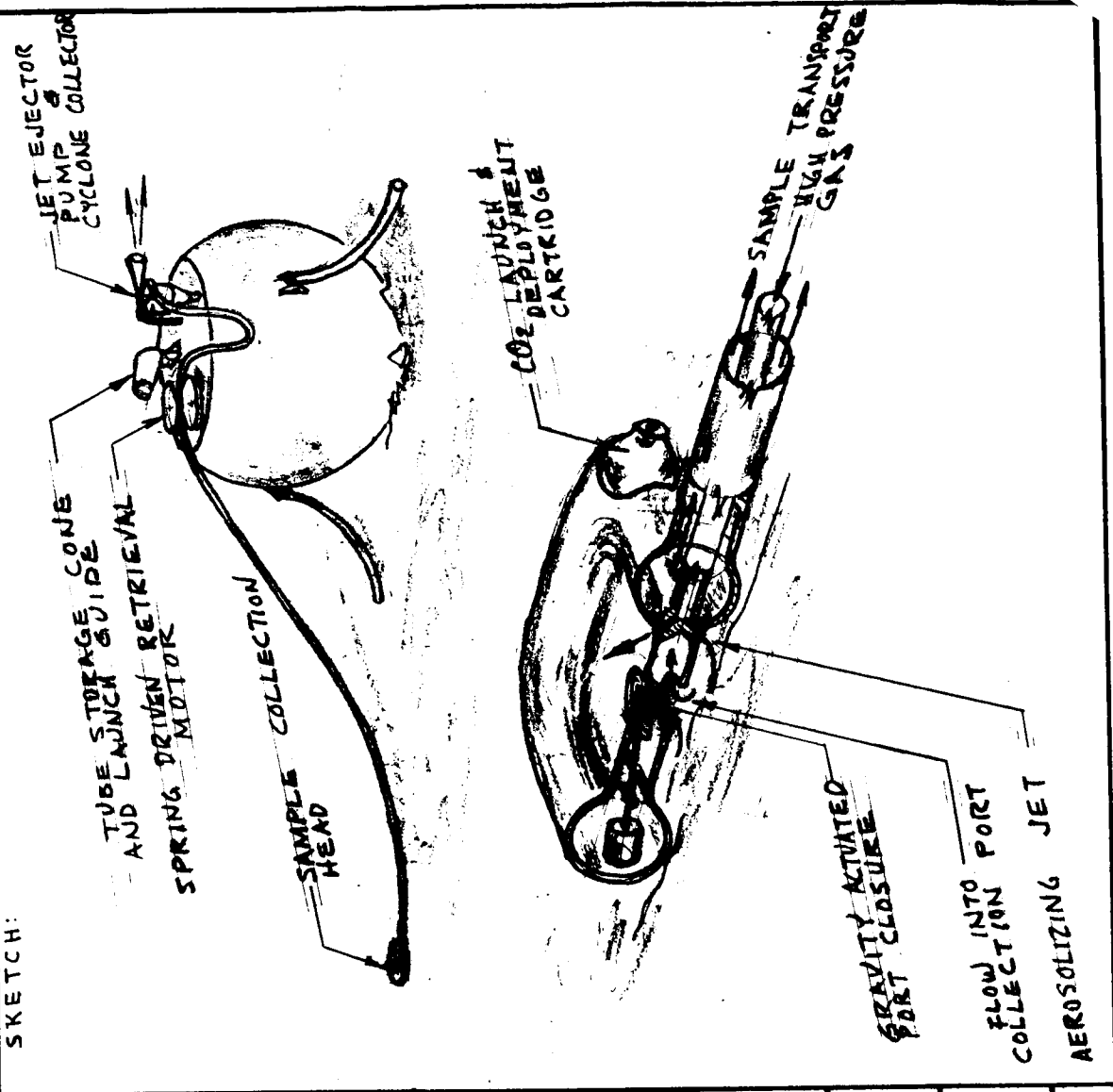
APPENDIX B

SAMPLING CONCEPTS

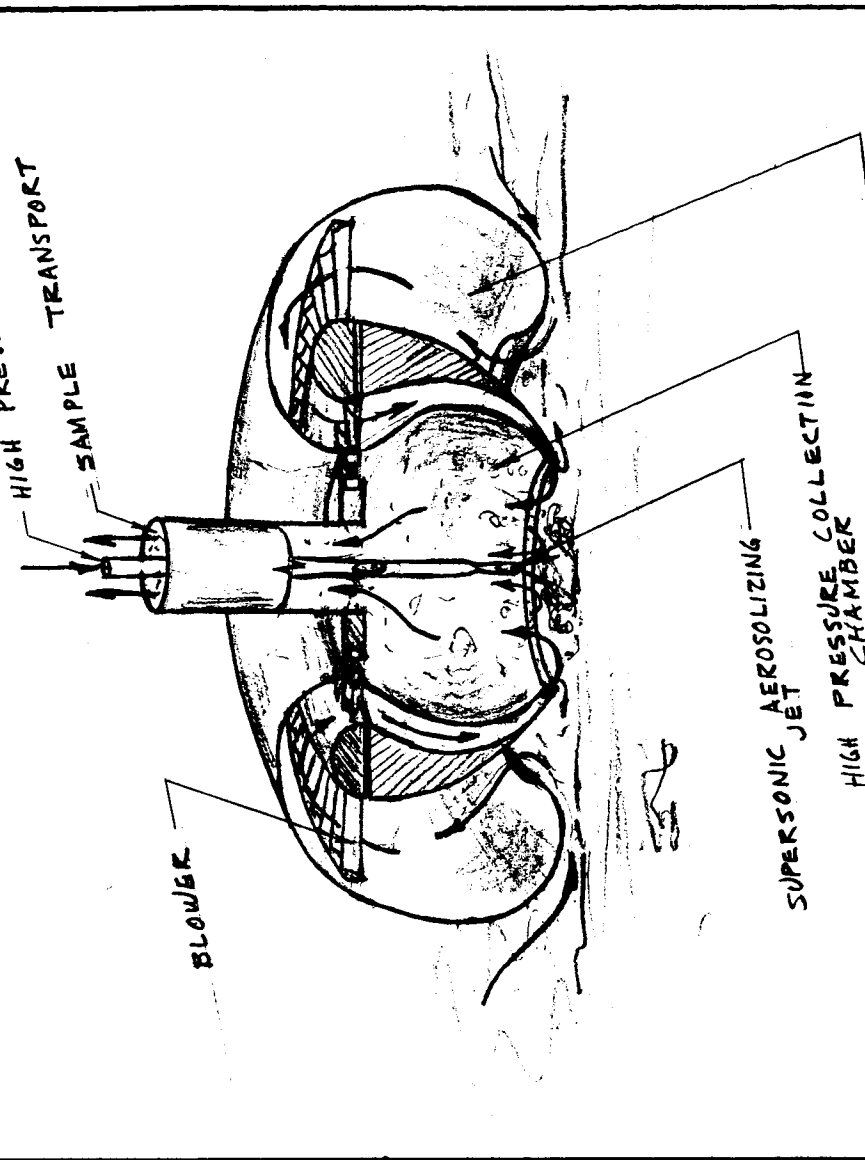
This appendix contains the thirty-two conceptual soil sampling mechanizations with which this program was initiated. Many of these concepts are obviously limited in their applications while several of those which were not developed may warrant further effort in the future.

The important point that was made clear in generating these concepts was that the methods of sample acquisition, sample transport into the payload, and sample processing are strongly interrelated. The ultimate development of each concept must consider these factors simultaneously throughout the design phase.

SAMPLING CONCEPT 1

<p>DESCRIPTIVE TITLE: AEROSOLIZING JET</p>	<p>SKETCH:</p> 
<p>SUPPORTING REQUIREMENTS: DEPLOYMENT: REACTION JETS PROVIDE BY TWO CO₂ CARTRIDGES, RANGE 10-15 FEET. SAMPLE TRANSPORT: PNEUMATIC TRANSPORT INSIDE FLEXIBLE TUBE SAMPLE PROCESSING: BY ACTION OF AEROSOL JET,</p>	
<p>OPERATIONAL CHARACTERISTICS: POWER: SIMPLICITY: MODERATE EST. WGT: 5 LBS EST. VOL: 800 IN³ THE SAMPLER IS DEPLOYED BY THE CO₂ SIMULTANEOUSLY PUNCTURING THE CARTRIDGE, PRODUCING REACTION JETS. THE SPRING DRIVEN MOTOR TURNS THE TAKE WHEELS REQUIRING 10 MINUTES TO RETRIEVE THE SAMPLER. THE SAMPLER COLLECTS LOOSE SOIL LIFTED BY THE AEROSOLIZING JETS. FLOW IS INDUCED THROUGH THE INTAKE PORT BY THE EJECTOR PUMP MOUNTED ON THE CYCLONE COLLECTOR. SOIL PARTICLES ARE CARRIED THROUGH THE TUBE WITH THIS FLOW. THE PICKUP HEAD WILL OPERATE IN EITHER OF TWO POSSIBLE POSITIONS. GRAVITY ACTUATED VALVE CLOSURES UNUSED PORT</p>	
<p>QUALITY OF SAMPLE: SAMPLE SIZE: ~ 5 GRAMS SAMPLE GRADE: AS 1MM SAMPLE DIAS: LOW DENSITY MATERIAL-SMALL SIZE SAMPLE DEGRADATION: NONE</p>	
<p>OVERALL PERFORMANCE: SOIL MODELS: ANY WITH LOOSE MATERIAL TERRAIN MODELS: ENVIRONMENTAL:</p>	
<p>REFERENCE DATA: LITTON INDUSTRIES STUDY</p>	

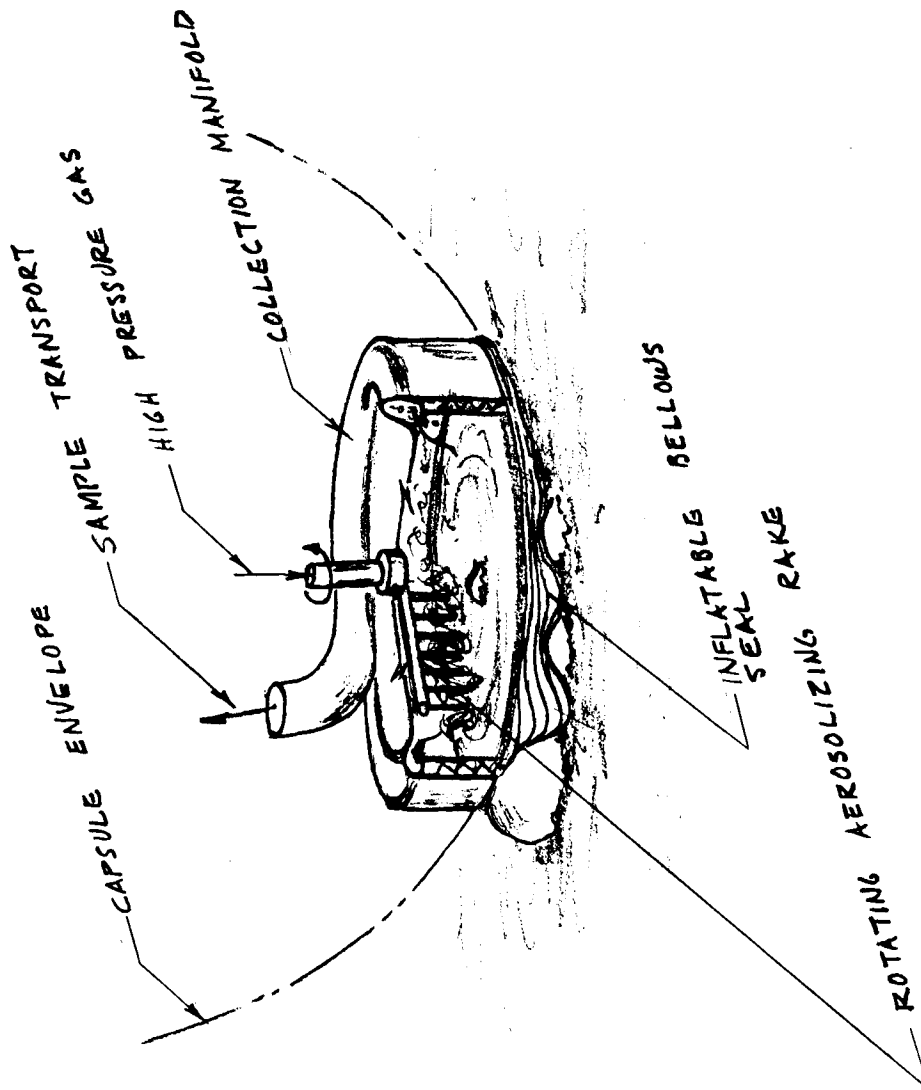
SAMPLING CONCEPT 2

DESCRIPTIVE TITLE: FORCE BALANCE PNEUMATIC COLLECTOR	SKETCH: 
SUPPORTING REQUIREMENTS: DEPLOYMENT: BOOM- RANGE 5-10 FEET	
SAMPLE TRANSPORT: PNEUMATIC	
SAMPLE PROCESSING: PNEUMATIC	
OPERATIONAL CHARACTERISTICS: POWER: HIGH EST WGT: EST. VOL: A BLOWER IS USED TO REDUCE THE PRESSURE IN THE OUTER CHAMBER AND INCREASE IT IN THE INNER CHAMBER. THE INTENT IS TO REDUCE THE NET LIFT FORCE TO ZERO OR LESS. SUPERSONIC JETS ARE USED IN THE INNER CHAMBER TO ERODE THE SOIL SURFACE. THE ROUNDED SHAPE WILL ALLOW THE SAMPLER TO RISE OVER LOCAL ROUGHNESS. A SIMPLER VERSION OF THIS APPROACH IS TO ELIMINATE THE BLOWER AND OUTER CHAMBER. FLOW THROUGH THE INNER ANNULAR SKIRT WHICH BECOMES THE OUTER SKIRT IS FROM A STORED GAS SUPPLY. LIFT FORCES ARE REACTED BY THE BOOM.	
QUALITY OF SAMPLE: SAMPLE SIZE: GRAMS SAMPLE GRADE: SAMPLE DIAS: LOOSE MATERIAL SAMPLE DEGRADATION:	
OVERALL PERFORMANCE: SOIL MODELS: TERRAIN MODELS: SMOOTH SURFACES ENVIRONMENTAL:	
REFERENCE DATA:	DATE: 2/26/66

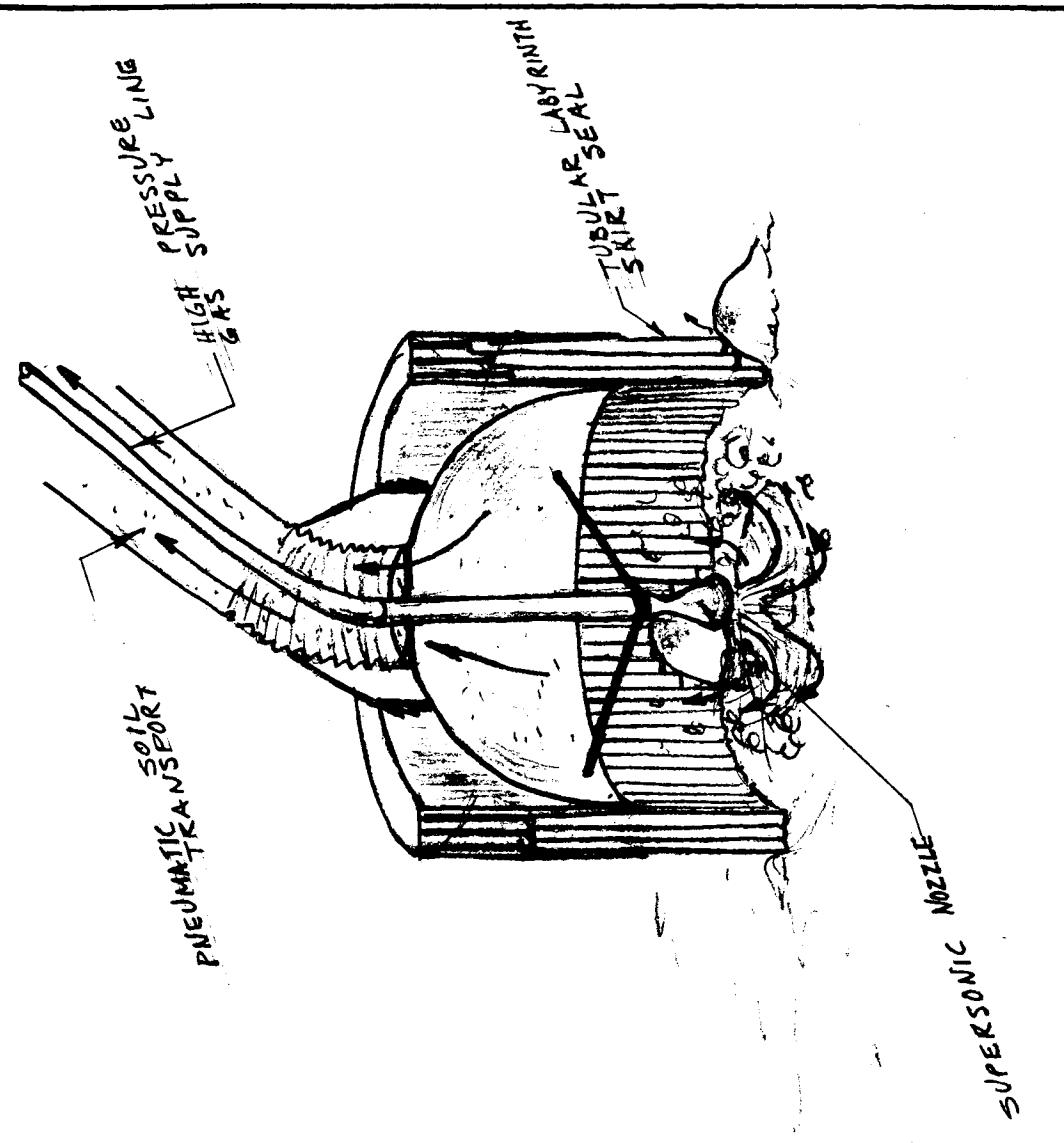
SAMPLING CONCEPT 3

<p>DESCRIPTIVE TITLE: SUPERPRESSURE PNEUMATIC WITH VERTICALLY DEPLOYED SEAL</p>	<p>SUPPORTING REQUIREMENTS:</p>
<p>DEPLOYMENT: INFLATION OF SEAL</p>	<p>SAMPLE TRANSPORT: PNEUMATIC</p>
<p>SAMPLE PROCESSING: PNEUMATIC DURING COLLECTION</p>	<p>OPERATIONAL CHARACTERISTICS:</p> <p>POWER: SIMPLICITY:</p> <p>EST WGT: EST VOL:</p> <p>AN ELASTOMERIC BELLOW SEAL IS INFLATED TO DEPLOY IT VERTICALLY TO THE SURFACE FROM THE BASE OF THE CAPSULE. THE AEROSOLIZING RAKE IS THEN DEPLOYED VERTICALLY AS IT IS ROTATED UNTIL IT REACHES THE SURFACE. VERTICAL FEED IS TERMINATED WHEN SOME MAXIMUM THRUST IS EXPERIENCED. THE TUBULAR RAKE TEETH ALSO FORM THE AEROSOLIZING JETS TO LIFT PARTICLES INTO SUSPENSION IN THE GAS FLOW, WHICH PASSES INTO THE COLLECTION MANFOLD TO THE TRANSPORT TUBE.</p>
<p>QUALITY OF SAMPLE:</p>	<p>SAMPLE GRADE:</p>
<p>SAMPLE SIZE:</p>	<p>SAMPLE DIAS:</p>
<p>SAMPLE DEGRADATION:</p>	<p>OVERALL PERFORMANCE:</p>
<p>SOIL MODELS:</p>	<p>TERRAIN MODELS:</p>
<p>ENVIRONMENTAL:</p>	<p>REFERENCE DATA: ABL STUDY</p>
<p>PAGE: 2/26/66</p>	

SKETCH:



SAMPLING CONCEPT 4

<p>DESCRIPTIVE TITLE: SUPERSONIC JET WITH A LABYRINTH SEAL</p>	<p>SKETCH:</p> 
<p>SUPPORTING REQUIREMENTS: DEPLOYMENT: BOOM OR VERTICAL</p>	
<p>SAMPLE TRANSPORT: PNEUMATIC</p>	
<p>SAMPLE PROCESSING: PERFORMED DURING COLLECTION BY PNEUMATIC GRADING.</p>	
<p>OPERATIONAL CHARACTERISTICS: POWER: SIMPLICITY: MODERATE EST. WGT: EST. VOL: A CLOSE PACKED TUBULAR ARRAY ON THE PERIPHERY OF THE SAMPLER FORMS A LABYRINTH SEAL WHICH CONFORMS TO THE SURFACE CONTOURS. A SUPERSONIC NOZZLE IMPINGES A HIGH VELOCITY JET ON THE SURFACE TO DISLodge SOIL PARTICLES AND GET THEM AIR BORNE. SAMPLE TRANSPORT IS BY PNEUMATIC FLOW IN THE RETURN TUBE. SECONDARY SAMPLE COLLECTION CAN BE MADE BY RECOVERING SOIL IMPACTED IN THE LABYRINTH SEAL TUBULAR ARRAY.</p>	
<p>QUALITY OF SAMPLE: SAMPLE SIZE: GRAMS SAMPLE GRADE: SAMPLE DIAS: SAMPLE DEGRADATION:</p>	
<p>OVERALL PERFORMANCE: SOIL MODELS: ALL MODELS EXCEPT BARE ROCK TERRAIN MODELS: ALL MODELS. ENVIRONMENTAL:</p>	
<p>REFERENCE DATA: LITTON DATA DATE: 2/26/66</p>	

SAMPLING CONCEPT 5

DESCRIPTIVE TITLE: DRAG LINE SCOOP, BRUSH, AND AEROSOLIZING SAMPLE (J-BAND, TYPEA)

SUPPORTING REQUIREMENTS:

DEPLOYMENT: BALLISTIC LAUNCH USING SPRINGS OR PNEUMATIC SYSTEM

SAMPLE TRANSPORT: MECHANICAL IN SMALL COLLECTION CHAMBER WHICH CAN BE SEPARATED FROM SAMPLER BODY

SAMPLE PROCESSING:

OPERATIONAL CHARACTERISTICS:

POWER: 25-50 WATTS SIMPLICITY: COMPLEX
EST WGT: SAMPLER 1LB, EST VOL: 20 IN³
RETRIEVAL SYSTEM 1LB

THE SAMPLER IS BALLISTICALLY DEPLOYED TO A RANGE OF 25 FEET. THE BRUSHES UNFOLD DURING FLIGHT. AFTER LANDING THE INITIAL PULL ON WIRE RETRIEVAL LINE TRIPS GAS CYLINDER RELEASE INITIATING GAS FLOW THROUGH COLLECTION CHAMBER AS THE SAMPLER IS PULLED ALONG SURFACE THE BRUSHES SWEEP MATERIAL TOWARDS INTAKE SCOOPS WHICH BRINGS SOIL INTO PROXIMITY OF AEROSOL JET. THE COLLECTION CHAMBER BREAKS AWAY IF THE BODY IS LODGED OR AT THE PAYLOAD

QUALITY OF SAMPLE:

SAMPLE SIZE: 1 GRAM SAMPLE GRADE:

SAMPLE DIAS:

SAMPLE DEGRADATION:

OVERALL PERFORMANCE:

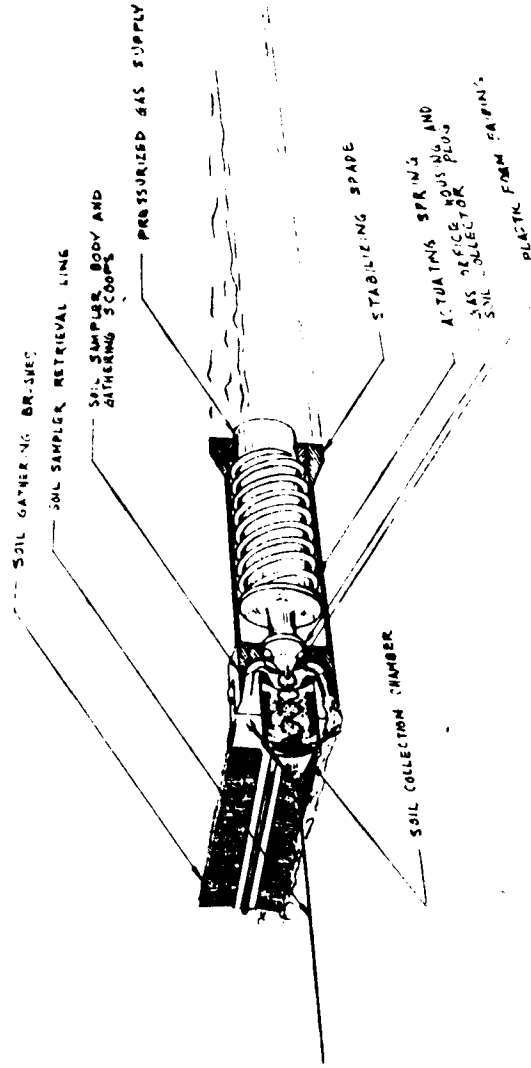
SOIL MODELS:

TERRAIN MODELS: FLAT OR SMOOTH SURFACES

ENVIRONMENTAL:

REFERENCE DATA: W.M. BACHLE - ORIGINAL CONCEPT
DEVELOPED 3/10/64 DATE: 2/25/66

SKETCH:

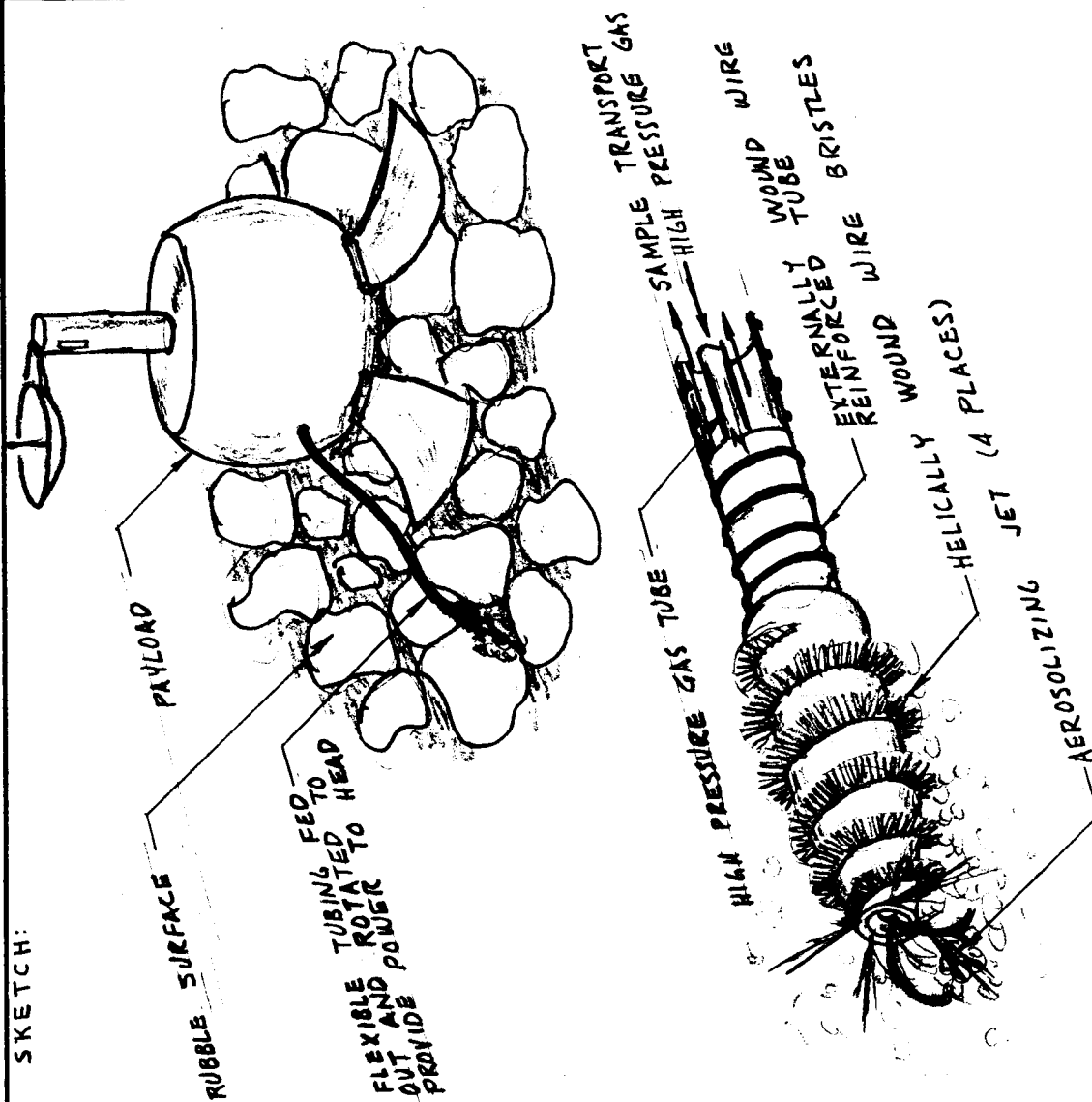


2D
(ALSO 3C)

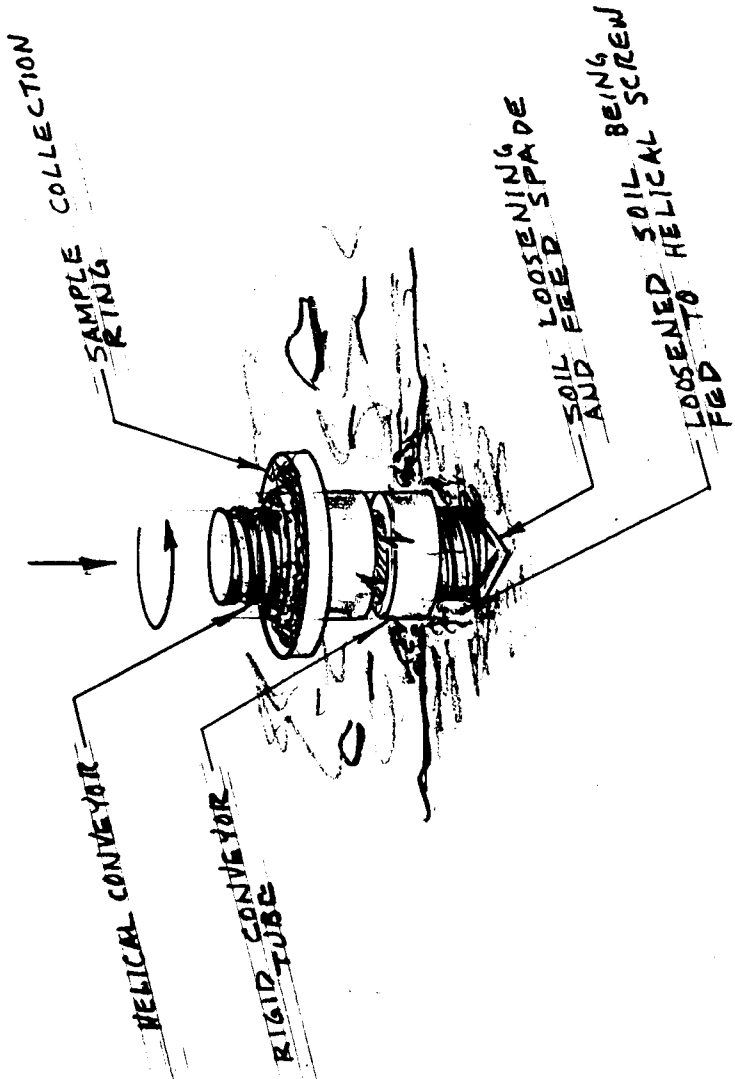
SAMPLING CONCEPT C

DESCRIPTIVE TITLE: BURROWING TUBE (TYPE A)	
SUPPORTING REQUIREMENTS: DEPLOYMENT: ROTARY FEED USING EXTERNAL WIRE REINFORCEMENT - MOTOR DRIVEN AT DEPLOYMENT PORT - DEPLOYED LENGTH 5-10 FEET SAMPLE TRANSPORT: PNEUMATIC TRANSPORT BY INDUCED FLOW IN TUBE SAMPLE PROCESSING: AUTOMATIC BY VELOCITY OF FLOW	
OPERATIONAL CHARACTERISTICS: POWER: 12-25 WATTS EST WGT: HEAD .5 TUBE 1.0 MOTOR .6 GAS SUPPLY 1.5 SIMPLICITY: MODERATE EST. VOL:	AS THE TUBE IS ROTATED & FED OUT FROM STORAGE, THE SPIRAL PATTERN OF THE BRISTLES ON THE HEAD PULLS IT ALONG THE SURFACE DISLODGING FINE SOIL PARTICLES. THE WEIGHT OF THE HEAD WILL TEND TO CAUSE IT TO SEEK THE LOWER LEVELS AND DESCEND INTO CRACKS AND CREVICES BETWEEN LARGE BOULDERS & COBBLES.
QUALITY OF SAMPLE: SAMPLE SIZE: GRAMS SAMPLE DIAS: UNKNOWN - PROBABLY SLIGHT SAMPLE DEGRADATION: NONE SAMPLE GRADE: d ≤ 500μ	
OVERALL PERFORMANCE: SOIL MODELS: WEAK & FINELY DISBURSED MATERIAL TERRAIN MODELS: ENVIRONMENTAL:	
REFERENCE DATA: T. W. NEUMANN LITERATURE STUDY FOR TRANSPORT	DATE: 2/21/66

SKETCH:



SAMPLING CONCEPT 7

<p>DESCRIPTIVE TITLE: RIGID HELICAL SCREW</p>	<p>SKETCH:</p> 
<p>SUPPORTING REQUIREMENTS: DEPLOYMENT: VERTICAL TO SURFACE</p>	
<p>SAMPLE TRANSPORT: MECHANICAL WITH HELICAL SCREW</p>	
<p>SAMPLE PROCESSING: CONTROLLED BY SCREW DIMENSIONS</p>	
<p>OPERATIONAL CHARACTERISTICS: POWER: EST. WGT: EST. VOL: SIMPLICITY: SIMPLE THE SCREW IS ROTATED TO LOOSEN SOIL AND GATHER IT AT THE EXPOSED END OF THE HELICAL WHERE IT IS PICKED UP AND CONVEYED TO THE TOP OF THE RIGID CONVEYOR TUBE AND COLLECTED IN THE SAMPLE COLLECTION RING. THE SOIL CAN BE TRANSFERRED FROM THE COLLECTION RING BY SPINNING ALTERNATIVELY THE RING CAN BE MADE TO SPLIT ALLOWING THE SAMPLE TO BE TRANSFERRED BY GRAVITY DROP.</p>	
<p>QUALITY OF SAMPLE: SAMPLE SIZE: 5-10 GRAMS SAMPLE GRADE: d 500µ SAMPLE DIAS: SAMPLE DEGRADATION: POSSIBLY DUE TO GRINDING</p>	
<p>OVERALL PERFORMANCE: SOIL MODELS: ALL BUT ROCK TERRAIN MODELS: ENVIRONMENTAL:</p>	
<p>REFERENCE DATA: JPL CONCEPT DATE: 3/18/66</p>	

SAMPLING CONCEPT 8

DESCRIPTIVE TITLE: FLEXIBLE HELICAL SCREW

SUPPORTING REQUIREMENTS:

DEPLOYMENT: THE FLEXIBLE HELICAL CONVEYOR IS FED OUT OF THE CAPSULE. TO ALLOW THE HEAD TO ADVANCE, THE FLUTES ON THE HEAD ASSIST IN DEPLOYMENT. SAMPLE TRANSPORT: MECHANICAL WITH HELICAL SCREW.

SAMPLE PROCESSING: CONTROLLED BY SCREW DIMENSIONS

OPERATIONAL CHARACTERISTICS:

POWER: EST. WGT: SIMPLICITY: SIMPLE
EST. VOL: THE SCREW AND HEAD ARE ROTATED TOGETHER WHILE THE FLEXIBLE CONVEYOR IS FED OUT OF THE CAPSULE. THE FLUTES OF THE HEAD FEEDS SOIL BACK TO THE EXPOSED HELICAL SCREW WHICH THEN CONVEYS IT BACK TO THE CAPSULE.

QUALITY OF SAMPLE:

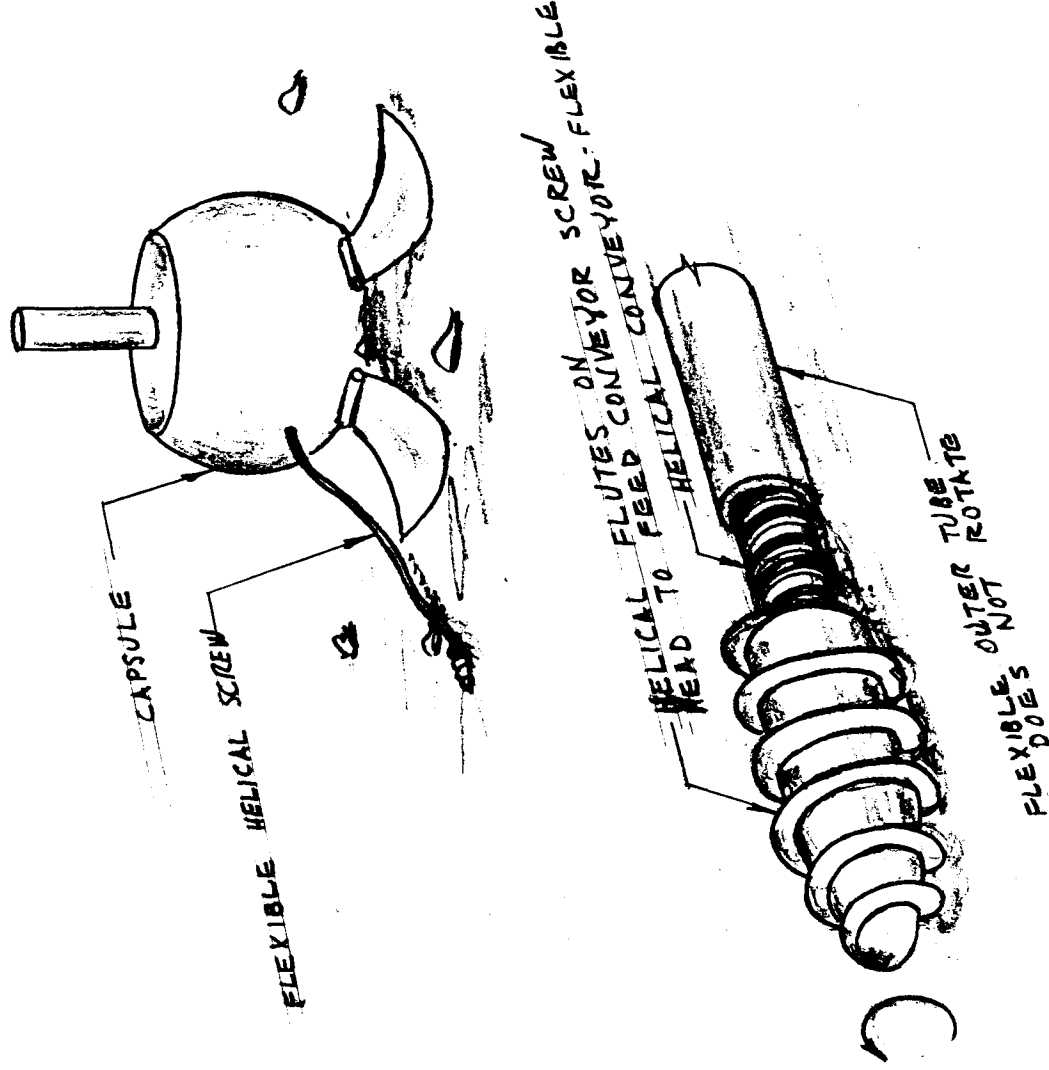
SAMPLE SIZE: < 5 GRAMS
SAMPLE GRADE: AS 5004
SAMPLE DIAS: POSSIBLY DUE TO GRINDING
SAMPLE DEGRADATION:

OVERALL PERFORMANCE:

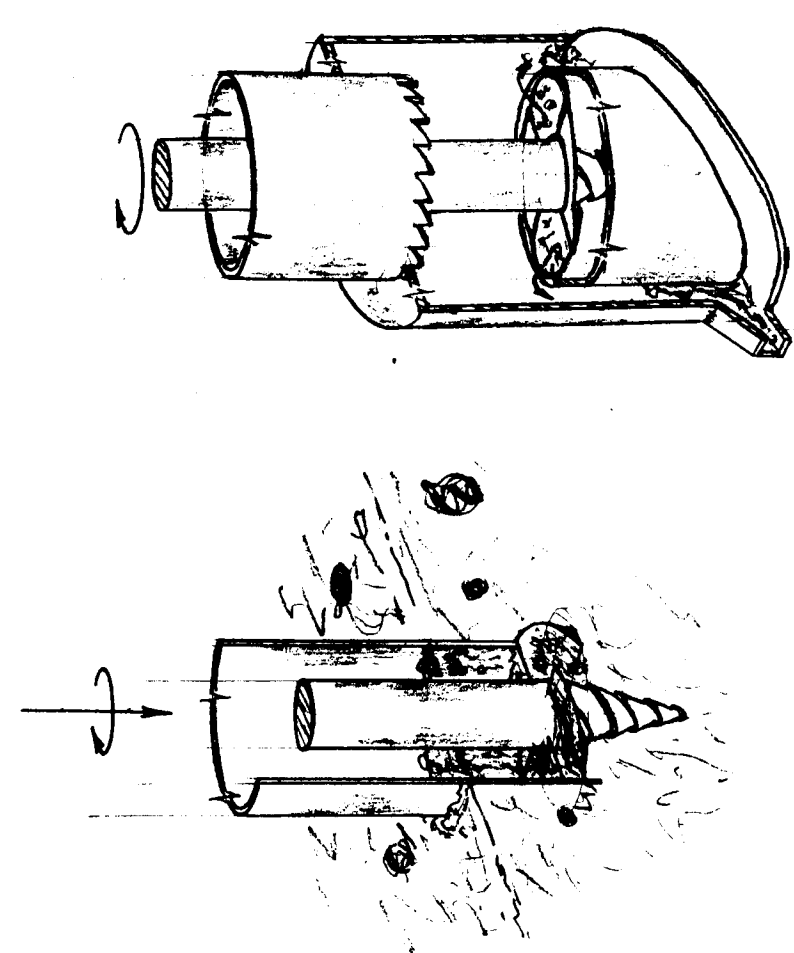
SOIL MODELS: LOOSE
TERRAIN MODELS: AGGREGATE
ENVIRONMENTAL:

REFERENCE DATA: JPL CONCEPT
DATE: 3/18/66

SKETCH:



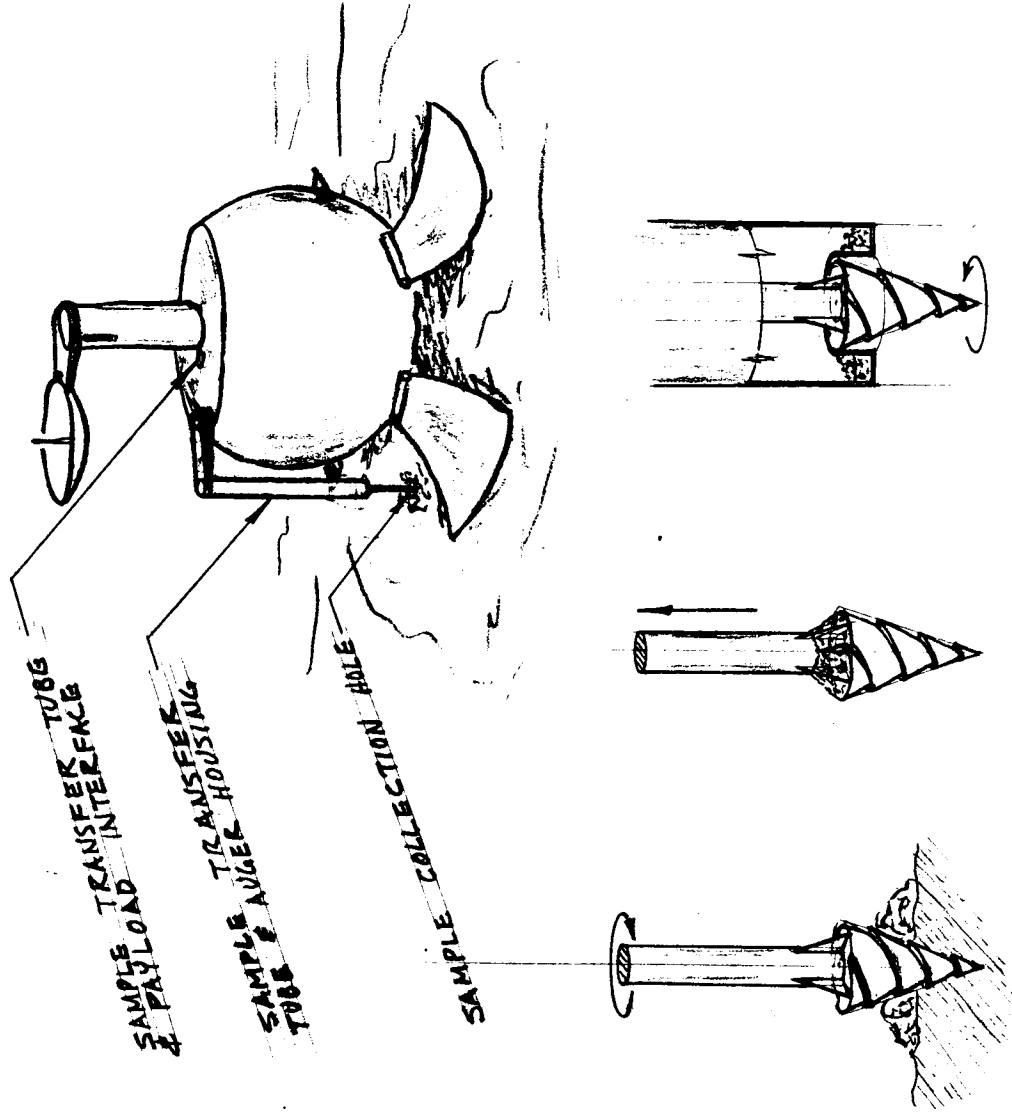
SAMPLING CONCEPT 9

DESCRIPTIVE TITLE: AUGER SAMPLER (TYPE B)	SKETCH:	
SUPPORTING REQUIREMENTS: DEPLOYMENT: VERTICAL - SCREW FEED		
SAMPLE TRANSPORT: MECHANICAL IN AUGER CASING - SPIN OFF INTO PROCESSING CHUTE - VIBRATION AND GRAVITY IN PROCESSING CHUTE	CONFIGURATION DURING SAMPLE COLLECTION	SAMPLE DUMP CONFIGURATION
SAMPLE PROCESSING: MAY BE DONE WITH INCLINED HELICAL SIEVES IN DUMP CHUTE		
OPERATIONAL CHARACTERISTICS: POWER: 25-100 WATTS SIMPLICITY: SIMPLE EST WGT: < 5 LBS EST VOL: 20-50 IN ³ THE AUGER IS ROTATED AND FED VERTICALLY INTO THE SURFACE. THE TRAILING EDGES OF THE AUGER FLUTES ARE FLEXIBLE AND SPRING LOADED TO REST ON THE TRAILING FLUTE. AS SOIL IS CUT FROM THE SURFACE THE FLEXIBLE FLUTES DEFELECT UP ALLOWING THE SOIL TO COLLECT IN THE CASING WHEN THE AUGER IS RETRACTED THE FLUTES RETURN TO THE CLOSED POSITION PREVENTING LOSS OF LOOSE GRANULAR MATERIAL. AFTER THE AUGER IS TRANSFERRED INTO THE PROCESSING OR DUMP CHAMBER THE CASING IS RETRACTED AND THE AUGER 'SPUN UP TO TRANSFER SOIL		
QUALITY OF SAMPLE: SAMPLE SIZE: GRAMS SAMPLE GRADE: d < 4mm SAMPLE DIAS: NONE SAMPLE DEGRADATION: NONE		
OVERALL PERFORMANCE: SOIL MODELS: ALL BUT SOLID ROCK OR RUBBLE TERRAIN MODELS: ENVIRONMENTAL:		
REFERENCE DATA: ABL STUDY	DATE: 2/23/66	

SAMPLING CONCEPT 10

DESCRIPTIVE TITLE: OPEN AUGER OR DRILL (TYPE B)	
SUPPORTING REQUIREMENTS: DEPLOYMENT: SHORT ROTATING BOOM DESIRABLE - MAY BE FIXED INSIDE PAYLOAD SO ONLY VERTICAL TRAVERSE IS REQUIRED SAMPLE TRANSPORT: MECHANICALLY FROM HOLE BY WITHDRAWING AUGER. SAMPLE TRANSFER BY SPIN-OFF TO TRANSFER TUBE, TRANSPORT TO PAYLOAD FROM TRANSFER TUBE BY GRAVITY DROP SAMPLE PROCESSING: BY DRILLING ACTION OF AUGER & SIZE OF COLLECTION SHOULDER	
OPERATIONAL CHARACTERISTICS: POWER: 25-100 WATTS SIMPLICITY: SIMPLE EST WGT: AUGER .5 EST. VOL: 20-50 IN. TRANSPORT TUBE .25 MOTOR .8 THE AUGER IS A CONICAL SHAPE OF SMALL DIAMETER WHICH IS DRILLED INTO ANY TYPE OF COHESIVE SOIL OR AGGREGATE. WHEN THE AUGER IS WITHDRAWN SOIL IS COLLECTED ON THE SHOULDER. MAXIMUM SIZE OF PARTICLE DETERMINED BY WIDTH OF SHOULDER. TOTAL SAMPLE SIZE DETERMINED BY SHOULDER WIDTH, NUMBER OF AUGER WITHDRAWALS, AND NUMBER OF SHOULDERS INCORPORATED ON AUGER SHAFT.	
QUALITY OF SAMPLE: SAMPLE SIZE: MGRNS - GRN SAMPLE GRADE: $d \leq 5 \text{ mm}$ SAMPLE DIAS: SAMPLE DEGRADATION:	
OVERALL PERFORMANCE: SOIL MODELS: ALL BUT SOLID ROCK TERRAIN MODELS: ENVIRONMENTAL:	
REFERENCE DATA: W.H. BACKLE	DATE: 2/21/66

SKETCH:



BORING TO
COLLECT SAMPLE

SAMPLE
RETURN

SAMPLE SPIN-OFF
INTO TRANSFER TUBE
OR DROP CHUTE

SAMPLING CONCEPT II

<p>DESCRIPTIVE TITLE: CORING AUGER</p>	<p>SKETCH:</p>	
<p>SUPPORTING REQUIREMENTS:</p>		
<p>DEPLOYMENT: VERTICAL TO THE SURFACE</p>		
<p>SAMPLE TRANSPORT: MECHANICAL IN CORING TUBE</p>		
<p>SAMPLE PROCESSING: NONE</p>		
<p>OPERATIONAL CHARACTERISTICS:</p>		
<p>POWER:</p>		
<p>EST WGT:</p>		
<p>EST VOL:</p>		
<p>SIMPLICITY: MODERATE</p>		
<p>THE EXTERNAL AUGER FLUTES BITE INTO THE SOIL TO PROVIDE THE THRUST NEEDED TO CARRY THE CORING TUBE BELOW THE SURFACE. THE FLEXIBLE FINGERS PROVIDE A MEANS OF BREAKING OFF THE CORE OR OF TRAPPING LOOSE GRANULAR MATERIAL. THE SAMPLE IS REMOVED FROM THE CORE TUBE WITH A PISTON WHICH DRIVES OUT BOTH THE SAMPLE AND THE FLEXIBLE FINGER TRAP TUBE. ALTERNATIVELY A SPLIT CORE TUBE MAY BE USED SO THAT IT CAN BE OPENED LONGITUDINALLY TO RELEASE THE SAMPLE.</p>		
<p>QUALITY OF SAMPLE:</p>		
<p>SAMPLE SIZE: 210 GRAMS</p>		
<p>SAMPLE GRADE: NATURAL</p>		
<p>SAMPLE DIAS: NONE</p>		
<p>SAMPLE DEGRADATION: NONE</p>		
<p>OVERALL PERFORMANCE:</p>		
<p>SOIL MODELS: ALL BUT ROCK</p>		
<p>TERRAIN MODELS:</p>		
<p>ENVIRONMENTAL:</p>		
<p>REFERENCE DATA:</p>	<p>DATE: 3/18/66</p>	

SAMPLING CONCEPT 12

DESCRIPTIVE TITLE: AUGER WITH INTERNAL SAMPLE BUCKET

SUPPORTING REQUIREMENTS:

DEPLOYMENT: VERTICALLY TO THE SURFACE

SAMPLE TRANSPORT: MECHANICAL IN SOIL COLLECTION CUP

SAMPLE PROCESSING: BY SIEVING

OPERATIONAL CHARACTERISTICS:

POWER: SIMPLICITY: MODERATE
EST WGT: EST. VOL:
AS THE EXTERNAL AUGER FLUTES BITE INTO THE SURFACE, THEY PROVIDE THE THRUST REQUIRED TO PENETRATE THE SOIL. LOOSEND SOIL SLIDES PAST THE SCREENED WINDOWS IN THE STEM THUS ALLOWING THE SMALLER PARTICLES TO FALL INSIDE INTO THE SAMPLE COLLECTION CUP. THIS CAN BE WITHDRAWN PERIODICALLY TO BAIL THE COLLECTED SAMPLE. THE SAMPLE CAN BE TRANSFERRED FROM THE CUP BY SPIN-OFF. ALTERNATIVELY THE CUP CAN BE DRAWN UP THE STEM LIKE A PISTON ALLOWING THE SOIL TO SPILL OVER THE SIDES AT THE TOP OF THE STEM. THE AUGER IS NOT REMOVED FROM GROUND.

QUALITY OF SAMPLE:

SAMPLE SIZE: GRAMS SAMPLE GRADE: $\approx 500\mu$
SAMPLE DIAS:
SAMPLE DEGRADATION: NONE

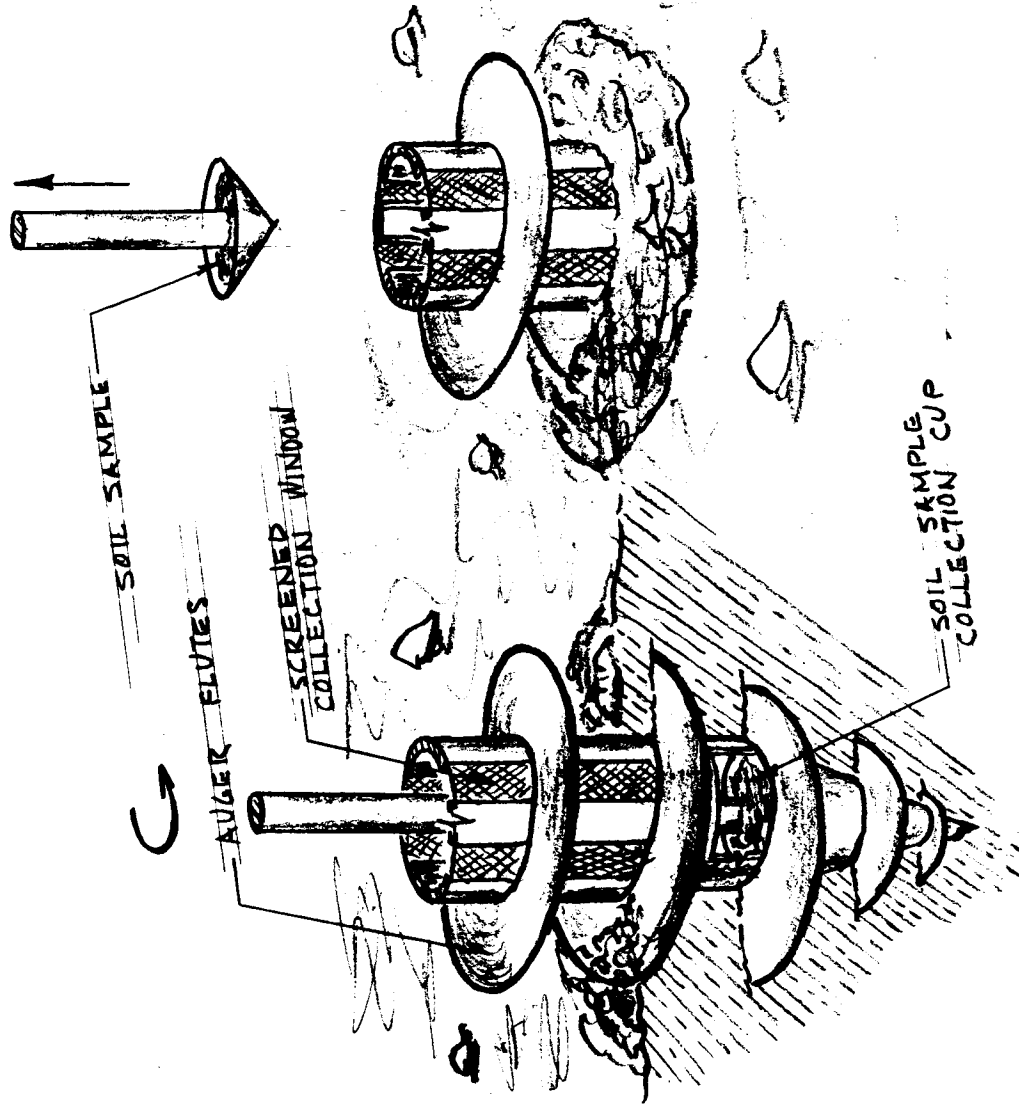
OVERALL PERFORMANCE:

SOIL MODELS: ALL BUT ROCK
TERRAIN MODELS: ALL
ENVIRONMENTAL:

REFERENCE DATA:

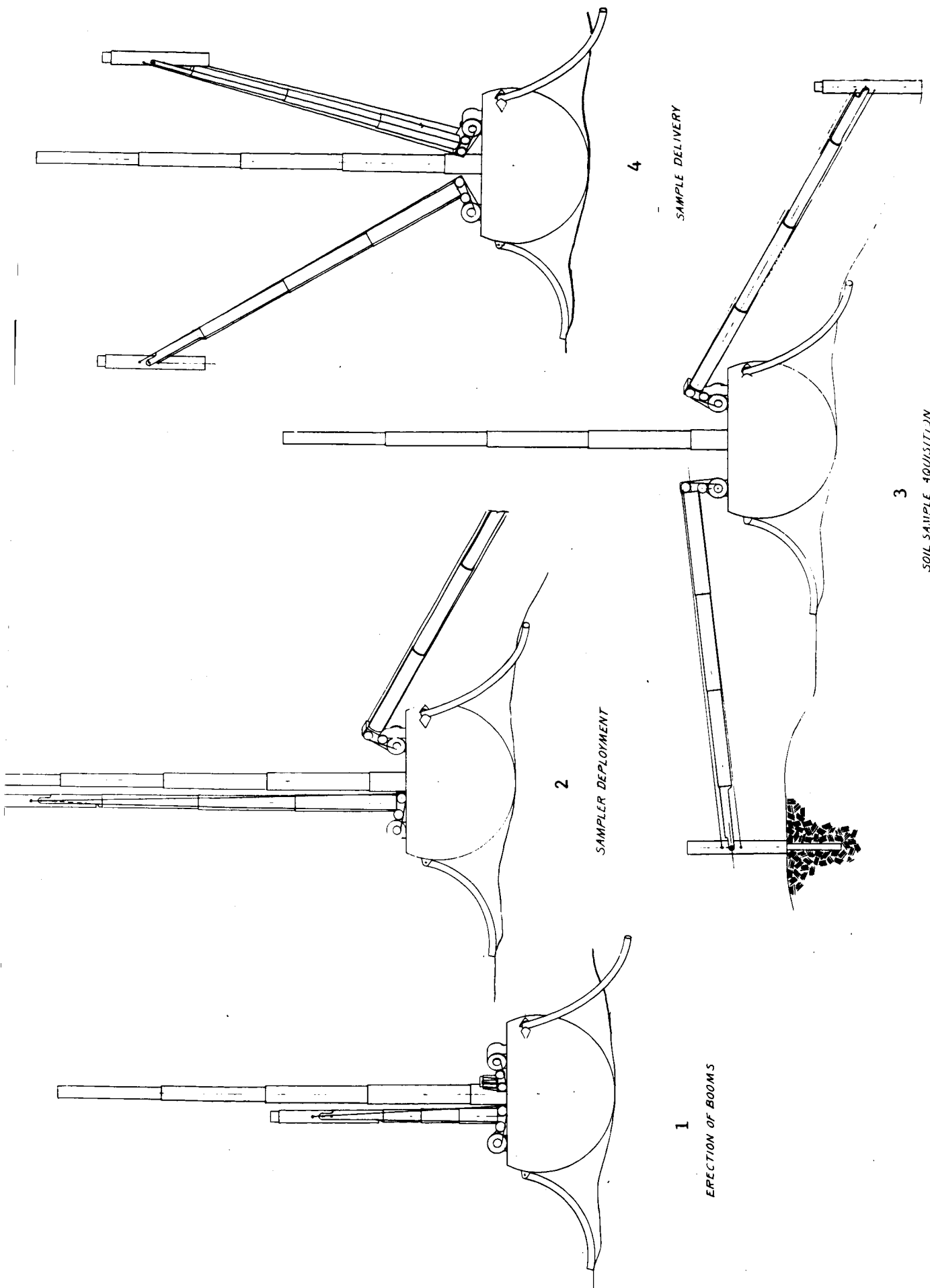
DATE: 3/18/66

SKETCH:



REFERENCE DATA: LUNAR SCIENCE PAYLOAD
STUDY (CONCEPT CONCEIVED 2/16/66) DATE: 2/25/66

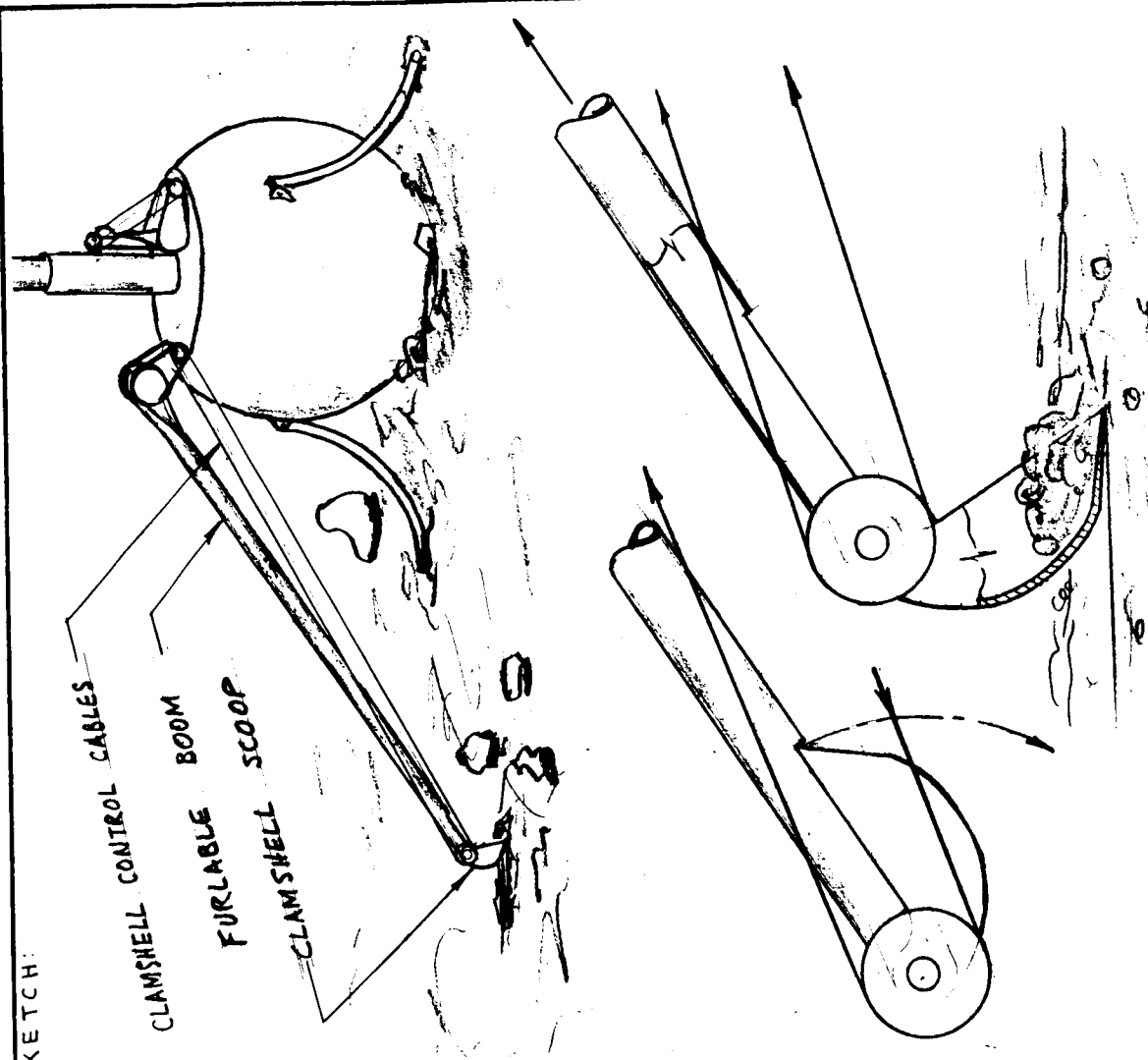




SAMPLING CONCEPT 14

DESCRIPTIVE TITLE: CLAM SHELL- FURLABLE BOOM	
SUPPORTING REQUIREMENTS: DEPLOYMENT: FURLABLE BOOM OR TELESCOPING BOOM - RANGE 5-10 FEET	
SAMPLE TRANSPORT: MECHANICAL IN SCOOP, GRAVITY DROP INSIDE BOOM TO CAPSULE	
SAMPLE PROCESSING: NONE- MAXIMUM SIZE PARTICLE COLLECTED DETERMINED BY WIDTH OF SCOOP	
OPERATIONAL CHARACTERISTICS: POWER: SLBS EST WGT: 5LBS SIMPLICITY: MODERATE EST. VOL:	
THE BOOM IS EXTENDED TO DESIRED LENGTH AND THE CLAM SHELL SCOOP OPENED BY DIFFERENTIAL MOTION OF CLAM SHELL CONTROL CABLES. THE SCOOP IS LOWERED TO THE SURFACE AND SOIL COLLECTED IS ACCOMPLISHED THROUGH COORDINATED RETRACTION OF BOTH THE BOOM AND THE CONTROL CABLES. THE SCOOP IS CLOSED BY DIFFERENTIAL MOVEMENT OF THE CABLES. THE SOIL SAMPLE IS DELIVERED THROUGH THE BOOM BY ERECTING TO VERTICAL USING GRAVITY DROP	
QUALITY OF SAMPLE: SAMPLE SIZE: GRAMS SAMPLE DIAS: NONE SAMPLE DEGRADATION: NONE	SAMPLE GRADE: COARSE
OVERALL PERFORMANCE: SOIL MODELS: COHESIVE & LOOSE SOILS TERRAIN MODELS: ALL BUT SOLID ROCK ENVIRONMENTAL	
REFERENCE DATA:	DATE: 3/9/66

SKETCH:



SAMPLING CONCEPT IS

DESCRIPTIVE TITLE: CLAM SHELL SAMPLER SCOOPS COMMAND ACTIVATED	SKETCH:
SUPPORTING REQUIREMENTS: DEPLOYMENT: BALLISTIC DEPLOYMENT 20-50 FEET	
SAMPLE TRANSPORT: MECHANICAL WITHIN SAMPLER BODY. PNEUMATIC FROM SAMPLER TO CAPSULE SAMPLE PROCESSING: PNEUMATIC DURING SAMPLE TRANSFER.	
OPERATIONAL CHARACTERISTICS: POWER: SIMPLICITY MODERATE EST WGT: EST VOL:	
ON COMMAND THE SCOOPS ARE DRIVEN OPEN TO COLLECT SOIL AS THE SAMPLER IS RETRIEVED. THE SCOOPS ARE CLOSED ON COMMAND OR USING A TIMED DELAY, AFTER RETRIEVAL THE SAMPLER IS DRAWN INTO A SOCKET AND SEALED, THE SAMPLE IS TRANSFERRED BY AEROSOLIZING WITHIN THE COLLECTOR WITH A HIGH VELOCITY JET AND ENTRAINING THE SOIL PARTICLES IN A FLOW.	
QUALITY OF SAMPLE: SAMPLE SIZE: GRAM SAMPLE GRADE: SAMPLE DIAS: SAMPLE DEGRADATION:	
OVERALL PERFORMANCE: SOIL MODELS: LOOSE AGGREGATE TERRAIN MODELS: ALL EXCEPT DESERT PAVEMENT ENVIRONMENTAL	
REFERENCE DATA:	
DATE: 3/8/66	

SAMPLING CONCEPT 16

DESCRIPTIVE TITLE: CLAM SHELL - SELF DEPLOYED

SUPPORTING REQUIREMENTS:

DEPLOYMENT: SELF DEPLOYED BY CRAWLING ACTION OF HEAD. RANGE UP TO 10-20 FEET

SAMPLE TRANSPORT: PNEUMATIC-CLOSED FLOW

SAMPLE PROCESSING: PERFORMED IN SEALED CLAMSHELL BY PNEUMATIC JETS AND TRANSPORT

OPERATIONAL CHARACTERISTICS:

POWER: SIMPLICITY: SIMPLE
EST WGT: < 5 LBS. EST. VOL:
THE CLAMSHELL HEAD IS LOWERED TO SURFACE FROM PAYLOAD. THE CLAMSHELL MOVES FORWARD BY EXTENDING THE FORWARD CLAMSHELL AND CONTRACTING TO PULL THE AFT CLAMSHELL FORWARD. SOIL IS COLLECTED IN THIS PROCESS AN AEROSOLIZING JET CAN ALSO BE INCORPORATED TO ASSIST IN COLLECTING THE SOIL. HIGH CLOSING FORCES CAN PROVIDE SHEARING AND CUTTING ACTION ON FIBROUS MATERIAL. SAMPLE GRADING IS ACCOMPLISHED INSIDE THE CLOSED CLAMSHELL - WHICH CAN BE PRESSURIZED - BY HIGH VELOCITY AIR JET. SAMPLE IS TRANSPORTED PNEUMATICALLY.

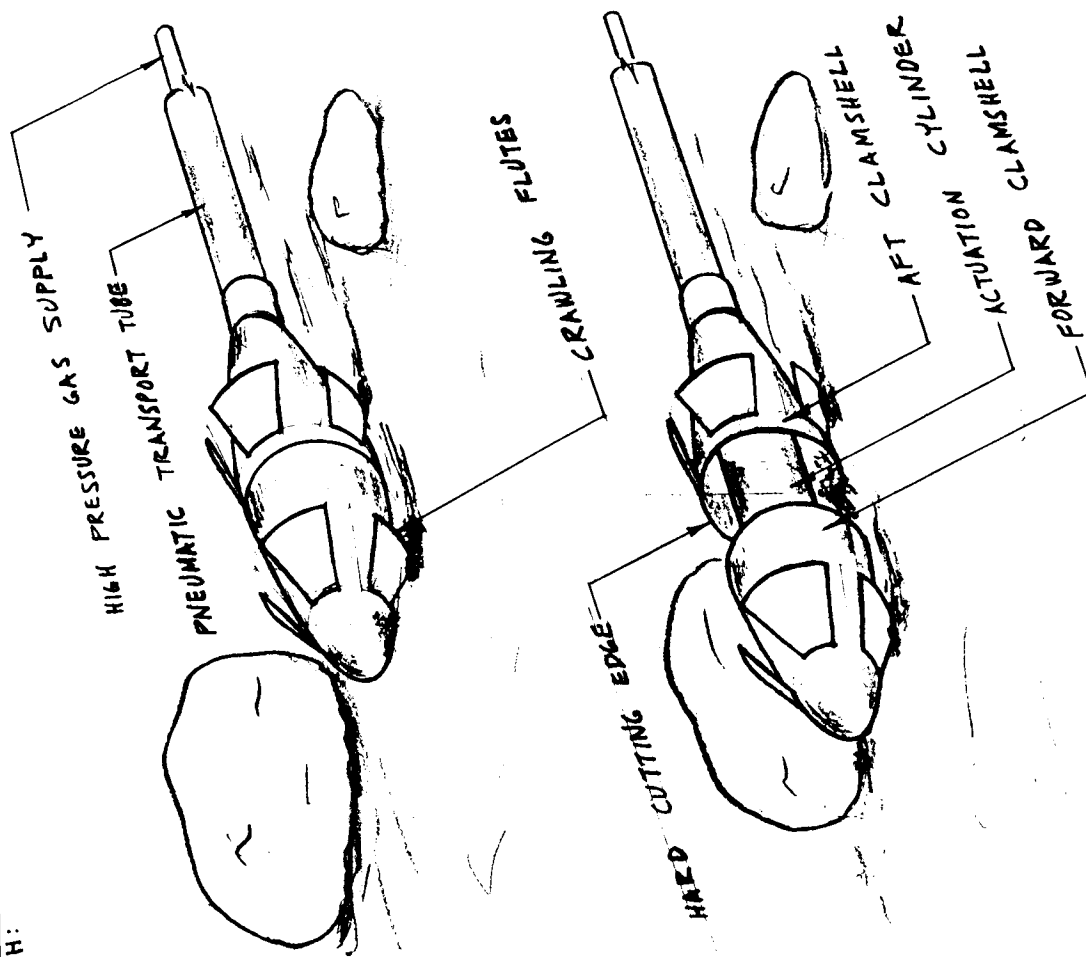
QUALITY OF SAMPLE: d ≤ 500μ
SAMPLE SIZE: GRAMS SAMPLE GRADE:
SAMPLE DIAS: NONE
SAMPLE DEGRADATION: NONE

OVERALL PERFORMANCE:
SOIL MODELS: ALL MODELS
TERRAIN MODELS: ALL MODELS
ENVIRONMENTAL:

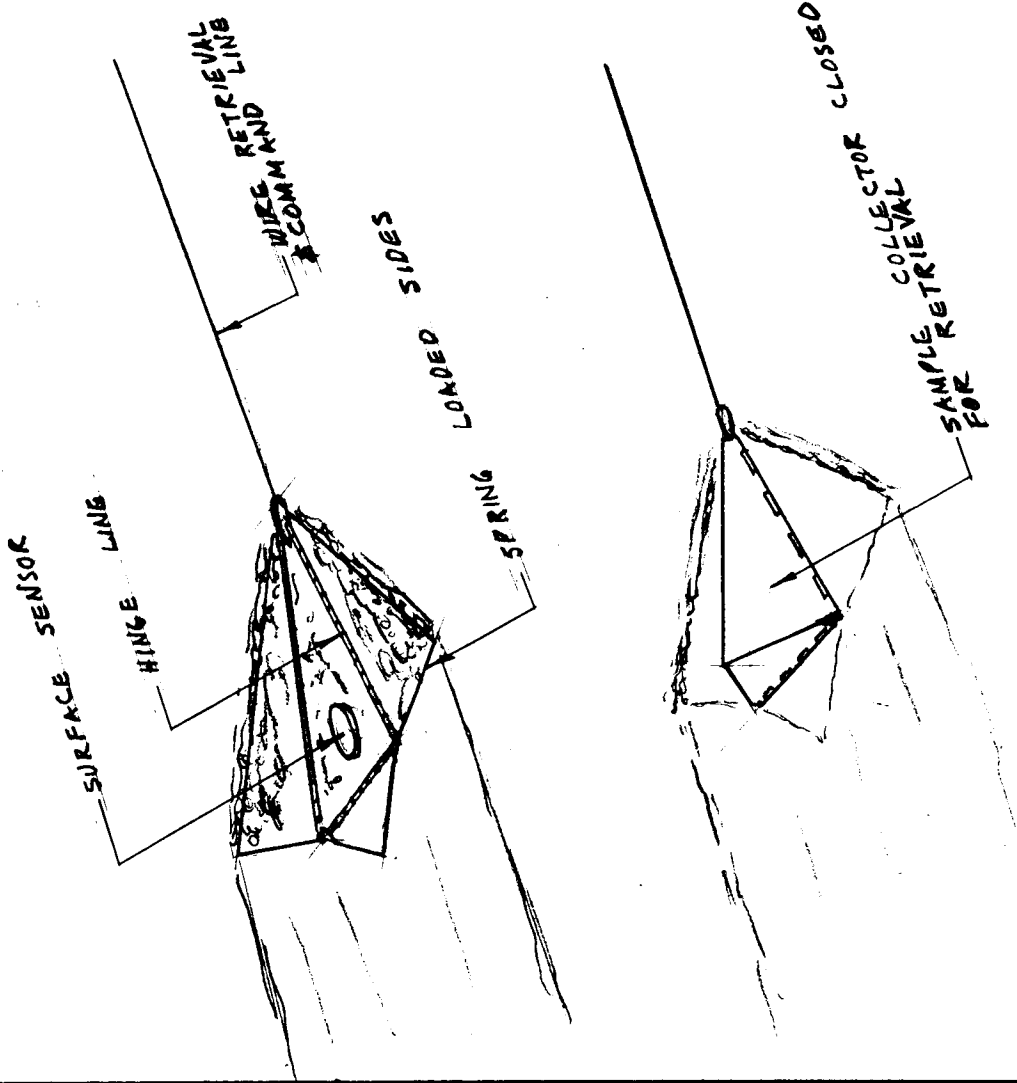
REFERENCE DATA: AERONAUTRONIC

DATE: 2/25/66

SKETCH:



SAMPLING CONCEPT 17

DESCRIPTIVE TITLE: SEMI-PASSIVE CLAMSHELL	SKETCH:
SUPPORTING REQUIREMENTS: DEPLOYMENT: BALLISTIC - RANGE 10-20 FEET	
SAMPLE TRANSPORT: MECHANICAL USING WIRE LINE RETRIEVAL.	
SAMPLE PROCESSING:	
OPERATIONAL CHARACTERISTICS: POWER: SIMPLICITY: SIMPLE EST WGT: EST. VOL:	
THE SAMPLER CONSISTS OF A TETRAHEDRON WITH HINGED SIDES, EACH SIDE IS SPRING LOADED WITH TWO SPRINGS SO THAT A FLAT SURFACE IS FORMED. THIS IS BALLISTICALLY DEPLOYED AND WILL LIE ON THE SURFACE IN ONE OF TWO PREFERRED ORIENTATIONS. SOIL IS GATHERED ON THE UPPER SURFACE EITHER BY WIND TRANSPORT OR BY TOWING THE SAMPLER ALONG THE SURFACE. THE SAMPLER IS CLOSED BY FUSING THE APPROPRIATE SPRING REACTION POINT ON ONE OF THE SPRINGS. THE REMAINING SPRING CLOSES THE BODY.	
QUALITY OF SAMPLE: SAMPLE SIZE: SAMPLE DIAS: SAMPLE DEGRADATION:	
OVERALL PERFORMANCE: SOIL MODELS: LOOSE AGGREGATE TERRAIN MODELS: RELATIVELY FLAT SURFACES ENVIRONMENTAL:	
REFERENCE DATA: AERONAUTRONIC DATE: 2/25/66	

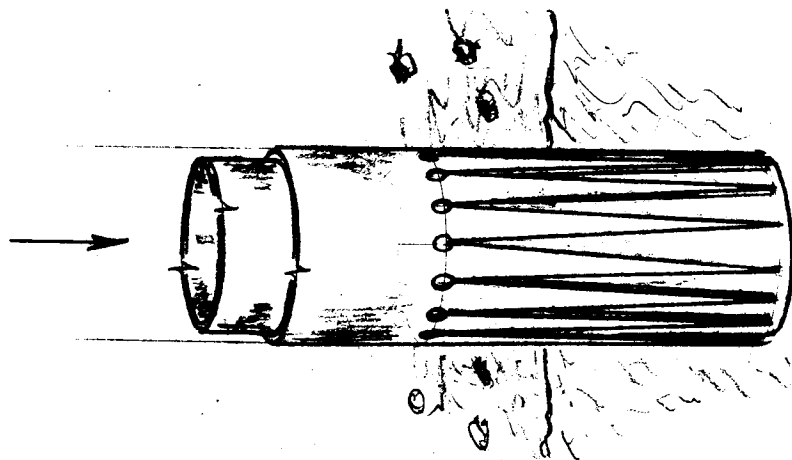
SAMPLING CONCEPT 18

<p>DESCRIPTIVE TITLE: ROTARY SCOOP</p>	<p>SKETCH:</p>	
<p>SUPPORTING REQUIREMENTS: DEPLOYMENT: VERTICAL EXTENSION ONLY IS REQUIRED</p> <p>SAMPLE TRANSPORT: MECHANICAL BY VIBRATING ACTION OF STEM AND INTERNAL CONFIGURATION.</p> <p>SAMPLE PROCESSING: BY CUTTING ACTION DURING COLLECTION AND BY SORTING ACTION OF TRANSPORT MOTION</p>		
<p>OPERATIONAL CHARACTERISTICS:</p> <p>POWER: EST. WGT: EST. VOL:</p> <p>SIMPLICITY: SIMPLE</p> <p>THE CONICAL TIP CONTAINS TWO EXTERNAL SCOOPS WHICH SHEARS THE SOIL AS THE STEM ROTATES. THE CENTRIFUGAL ACTION CARRIES THE SOIL TO THE WALL OF THE STEM WHERE THE VERTICAL VIBRATION CAUSES THE SOIL TO BE TRANSPORTED UP THE SIDES. THE SIZE OF THE LEDGES WILL DETERMINE THE MAXIMUM SIZE OF SOIL PARTICLE THAT IS TRANSPORTED. - IF LOW ROTATION SPEED IS DESIRED, A CONCENTRIC INNER STEM WILL BE REQUIRED TO ACHIEVE VERTICAL TRANSPORT OF THE SOIL.</p>		
<p>QUALITY OF SAMPLE:</p> <p>SAMPLE SIZE: 5-10 GRAMS</p> <p>SAMPLE DIAS: SORTED BY SIZE</p> <p>SAMPLE DEGRADATION:</p>		
<p>OVERALL PERFORMANCE:</p> <p>SOIL MODELS: ALL BUT SOLID ROCK</p> <p>TERRAIN MODELS:</p> <p>ENVIRONMENTAL:</p>		
<p>REFERENCE DATA:</p> <p>DATE: 3/17/66</p>		

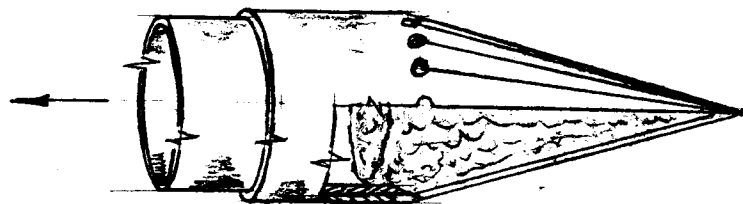
SAMPLING CONCEPT 19

DESCRIPTIVE TITLE: PLUG SAMPLER (TYPE B)
SUPPORTING REQUIREMENTS: DEPLOYMENT: VERTICALLY DEPLOYED BY MEANS OF EITHER HIGH THRUST OR HIGH VELOCITY IMPACT SAMPLE TRANSPORT: MECHANICAL BY VERTICAL RETRACTION FOLLOWED BY GRAVITY DROP COMBINED WITH MECHANICAL EXPULSION OF SAMPLE FROM TUBE. SAMPLE PROCESSING: NONE
OPERATIONAL CHARACTERISTICS: POWER: SIMPLICITY: SIMPLE EST WGT: < 5 LBS EST VOL: 20-50 IN ³ SAMPLER IS ACCELERATED PNEUMATICALLY KINETIC ENERGY OF TUBE CAUSES IT TO IMBED IN THE SOIL. THE INNER TUBE IS THEN RETRACTED ALLOWING THE FINGERS TO CLOSE TRAPPING A LOOSE AGGREGATE, IN COHESIVE SOIL A SOLID PLUG IS PROBABLY RETAINED. EXTRACTION CAN BE ACCOMPLISHED WITH A PNEUMATIC PISTON & CYLINDER. THE SAMPLE IS DUMPED BY EXTENDING THE INSIDE TUBE TO SPREAD THE FINGERS. A PNEUMATIC RAM CAN BE USE TO FORCE OUT A COHESIVE SAMPLE.
QUALITY OF SAMPLE: SAMPLE SIZE: GRAMS SAMPLE GRADE: UNDEFINED SAMPLE DIAS: NONE SAMPLE DEGRADATION: NONE
OVERALL PERFORMANCE: SOIL MODELS: ALL BUT LARGE GRAVEL & ROCK TERRAIN MODELS: ENVIRONMENTAL:
REFERENCE DATA: ABL STUDY DATE: 2/23/66

SKETCH:



DEPLOYMENT CONFIGURATION



RETRIEVAL CONFIGURATION

SAMPLING CONCEPT 20

DESCRIPTIVE TITLE: FINE TUBE PLUG ARRAY

SUPPORTING REQUIREMENTS:
DEPLOYMENT: VERTICAL

SAMPLE TRANSPORT: MECHANICAL IN
SAMPLING TUBES.

SAMPLE PROCESSING: DETERMINED BY
TUBE SIZE

OPERATIONAL CHARACTERISTICS:

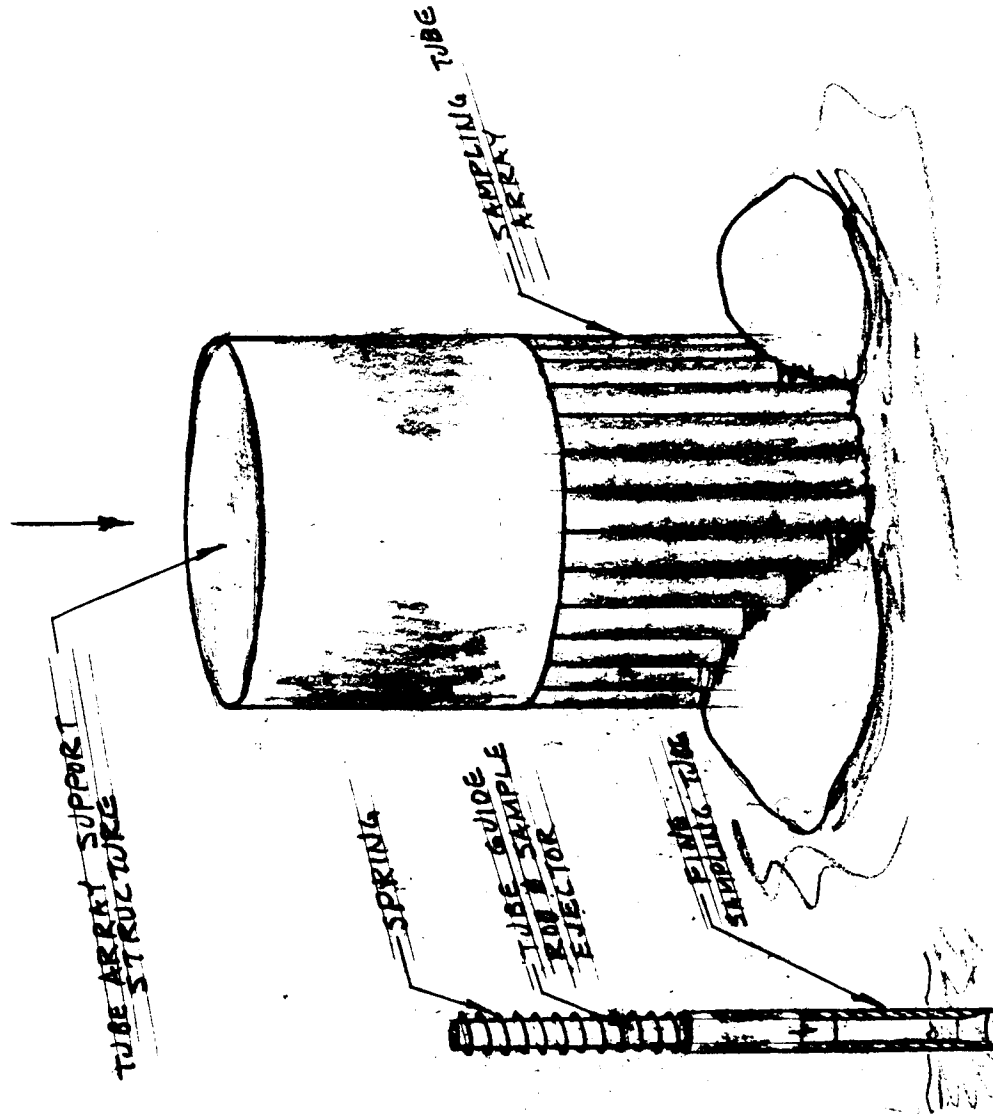
POWER: SIMPLICITY: MODERATE
EST WGT: EST VOL:
THE SAMPLER IS PRESSED VERTICALLY INTO THE SURFACE. TWOSE TUBES MEETING THE SURFACE. THOSE TUBES MEETING THE SURFACE RIDE UP THE GUIDE ROD AGAINST THE SPRING. THOSE TUBES PENETRATING THE SURFACE OF SURFACE SENSED TO GIVE A MAP OF SURFACE ROUGHNESS. THOSE TUBES PENETRATING THE SOIL COLLECT A SMALL SAMPLE WHICH IS PACKED INTO IT. THE SAMPLE IS REMOVED BY PRESSING THE SAMPLER ONTO A RIGID GRATING. FORCING THE GUIDE RODS DOWN INTO THE TUBE TO EJECT THE SOIL SAMPLE.

QUALITY OF SAMPLE:
SAMPLE SIZE: < 5 GRAMS
SAMPLE GRADE: d 51mm
SAMPLE DIAS: LOOSE OR COHESIVE AGGREGATE
SAMPLE DEGRADATION: NONE

OVERALL PERFORMANCE:
SOIL MODELS: LOOSE AGGREGATE OR WEAK COHESIVE
TERRAIN MODELS:
ENVIRONMENTAL:

REFERENCE DATA: DATE: 3/16/66

SKETCH:

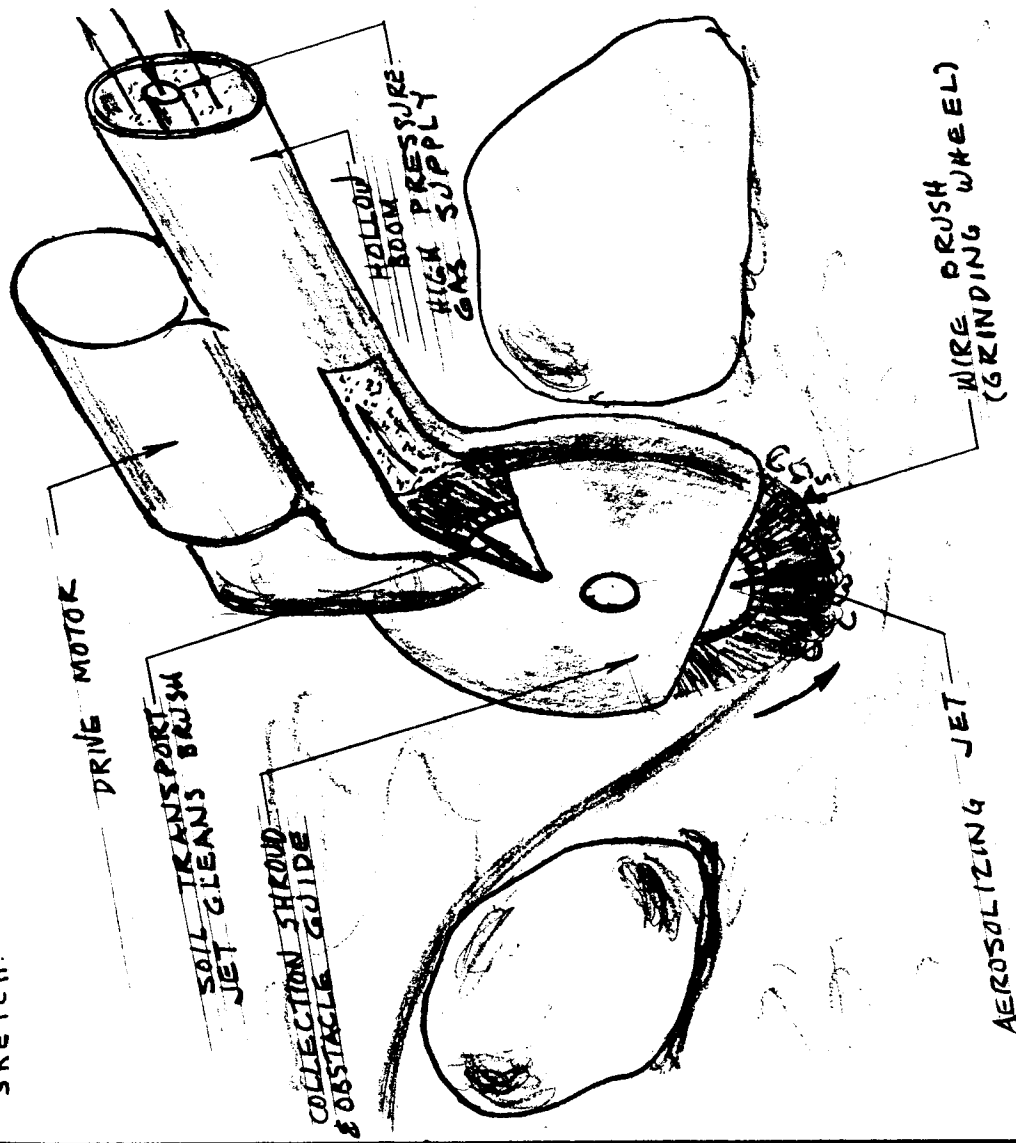


INDIVIDUAL TUBE
ELEMENT OF THE ARRAY

SAMPLING CONCEPT 21

<p>DESCRIPTIVE TITLE: ROTATING WIRE BRUSH WITH AEROSOLIZING JET</p>	
<p>SUPPORTING REQUIREMENTS:</p> <p>DEPLOYMENT: FURABLE BOOM OR TELESCOPING TUBE BOOM. RANGE 5-10 FEET</p>	
<p>SAMPLE TRANSPORT: MECHANICAL AND PNEUMATIC</p>	
<p>SAMPLE PROCESSING:</p>	
<p>OPERATIONAL CHARACTERISTICS:</p> <p>POWER: SIMPLICITY: MODERATE EST WGT: EST VOL:</p> <p>AS THE BOOM IS RETRACTED THE WIRE WHEEL ROTATES DISLODGING SOIL PARTICLES ASSISTED BY AN AEROSOLIZING JET. THE PARTICLES ARE CARRIED INTO THE HOOD BY THE WIRE WHEEL AS WELL AS THE INDUCED AIR FLOW INTO THE HOOD. AUXILIARY HIGH VELOCITY JETS INSIDE THE HOOD WILL BLOW SOIL PARTICLES OUT OF THE BRUSH INTO THE TRANSPORT TUBE. THE ROTATION OF THE WHEEL TENDS TO KEEP THE BOOM IN TENSION. THE HOOD WILL GUIDE THE WHEEL BETWEEN BOULDERS AS IT IS DRAWN TOWARDS THE CAPSULE</p>	
<p>QUALITY OF SAMPLE:</p> <p>SAMPLE SIZE: SAMPLE GRADE:</p> <p>SAMPLE DIAS: SAMPLE DEGRADATION:</p>	
<p>OVERALL PERFORMANCE:</p> <p>SOIL MODELS: ALL MODELS TERRAIN MODELS: ALL MODELS ENVIRONMENTAL:</p>	
<p>REFERENCE DATA:</p>	<p>DATE: 3/9/66</p>

SKETCH:

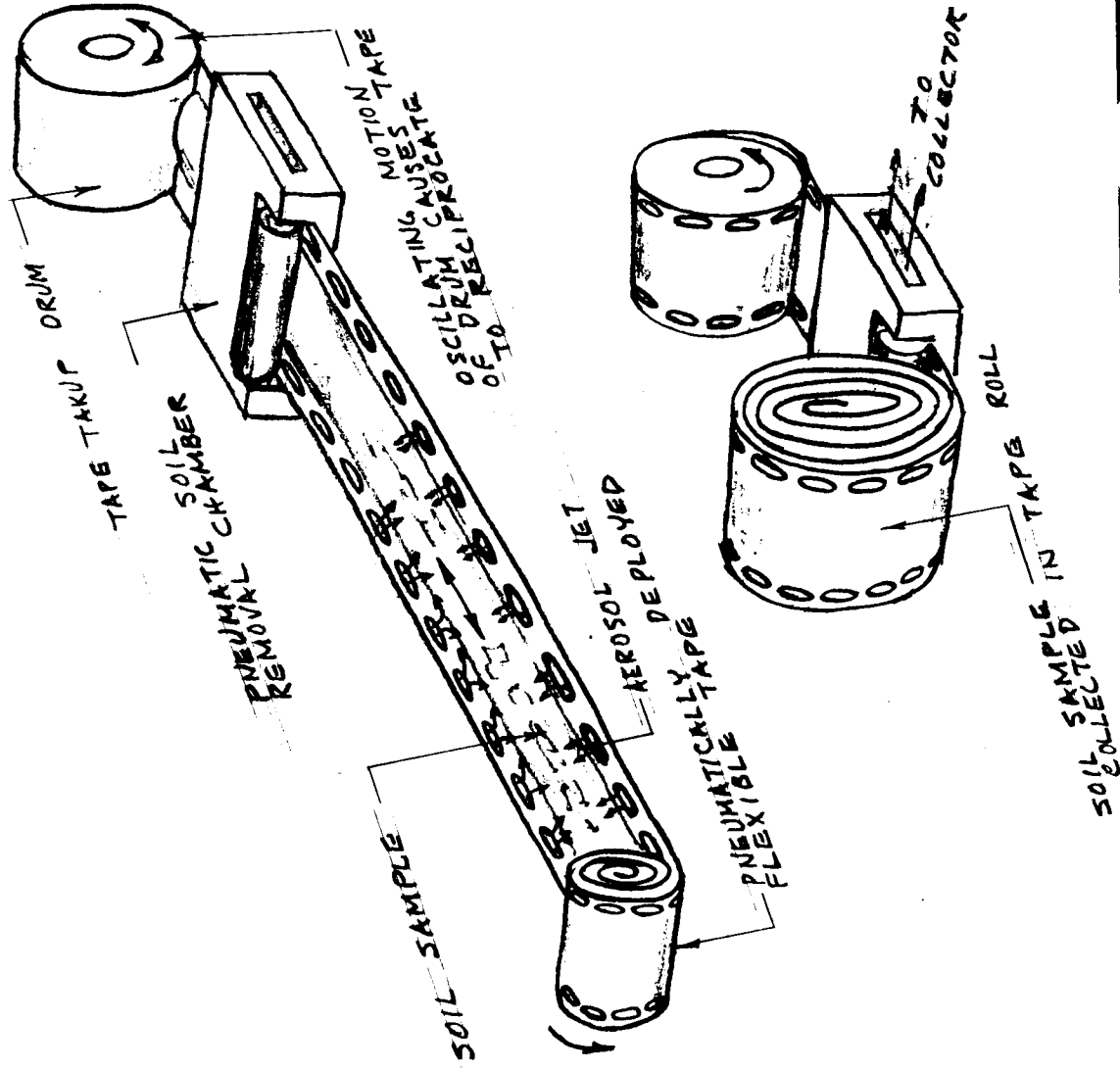


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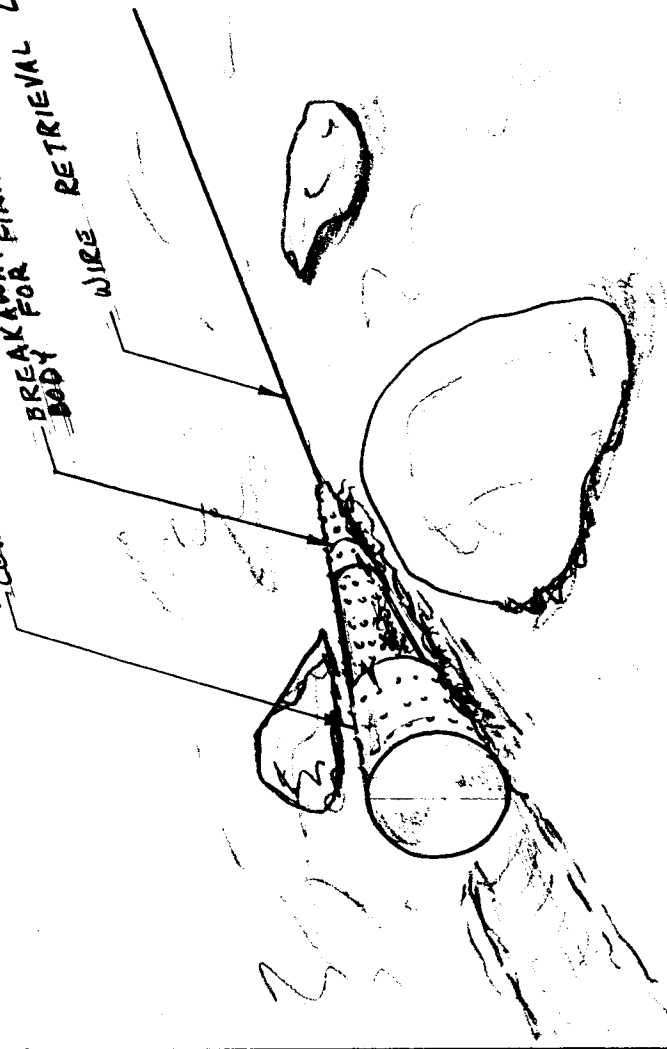
SAMPLING CONCEPT 22

DESCRIPTIVE TITLE: FLEXIBLE TAPE ABRADING SOIL SAMPLER	
SUPPORTING REQUIREMENTS: DEPLOYMENT: PNEUMATIC INFLATION OF TUBE CAUSES IT TO UNROLL ALONG THE SURFACE. RANGE 5 TO 15 FEET. SAMPLE TRANSPORT: REDUCING PRESSURE IN TAPE ALLOWS IT TO ROLL UP COLLECTING THE SOIL IN THE TAPE ROLL AS IT RETURNS TO THE CAPSULE. SAMPLE PROCESSING: PNEUMATIC AND MECHANICALLY BY SIZE OF SLOTS ALONG EDGE OF TAPE	
OPERATIONAL CHARACTERISTICS: POWER: EST WGT: SIMPLICITY: SIMPLE EST. VOL: WHEN THE TAPE HAS BEEN DEPLOYED TO ITS FULL LENGTH THE TAKEUP DRUM IS OSCILLATED BACK AND FORTH CAUSING THE TAPE TO ABRADE THE SURFACE WITH A RECIPROCATING MOTION. AS SOIL WORKS THROUGH THE EDGE SLOTS SMALL JETS BLOW IT TOWARD THE CENTER OF THE TAPE. RELASING THE PRESSURE ALLOWS THE TAPE TO RECOIL RETURNING THE COLLECTED SAMPLE TO THE CAPSULE. FINAL RETRIEVAL OF THE SAMPLE IS ACHIEVED BY TRANSFERRING THE TAPE ONTO THE TAKEUP DRUM THEREBY DRAWING IT THROUGH THE PNEUMATIC SOIL REMOVAL CHAMBER	
QUALITY OF SAMPLE: SAMPLE SIZE: < 5 GRMS SAMPLE DIAS: SAMPLE DEGRADATION: SAMPLE GRADE: d's SOIL	
OVERALL PERFORMANCE: SOIL MODELS: ALL MODELS TERRAIN MODELS: ALL EXCEPT VERY UNEVEN ENVIRONMENTAL:	
REFERENCE DATA: DATE: 3/15/66	

SKETCH:



SAMPLING CONCEPT 23

DESCRIPTIVE TITLE: ABRADING DRAG LINE	SKETCH:
SUPPORTING REQUIREMENTS: DEPLOYMENT: BALLISTIC DEPLOYMENT TO 10-50 FEET	
SAMPLE TRANSPORT: MECHANICAL INSIDE ABRADING BODY USING WIRE LINE RETRIEVAL - GRAVITY DUMP INSIDE CAPSULE	
SAMPLE PROCESSING: AUTOMATIC DURING COLLECTION	
OPERATIONAL CHARACTERISTICS: POWER: SIMPLICITY: SIMPLE EST WGT: EST VOL: THE SAMPLER IS CONICAL IN SHAPE WITH FORWARD FACING SMALL SCOOPS DISTRIBUTED OVER ABRASIVE COVERED SURFACE. AS THE SAMPLER IS RETRIEVED THE SURFACE AND PARTICLES ENTER SMALL SCOOPS, AND ARE COLLECTED INSIDE BODY AT THE RECEIVING SOCKET AND THE BREAKAWAY NOSE IS DETACHED, ALLOWING THE SOIL SAMPLE TO BE TRANSFERRED INTO THE CAPSULE BY GRAVITY.	
QUALITY OF SAMPLE: SAMPLE SIZE: SMALL SAMPLE GRADE: $d \leq 50\mu$ SAMPLE DIAS: SAMPLE DEGRADATION:	
OVERALL PERFORMANCE: SOIL MODELS: ALL BUT BARE ROCK TERRAIN MODELS: ALL BUT DESERT PAVEMENT ENVIRONMENTAL:	
REFERENCE DATA: DATE: 3/8/66	

SAMPLING CONCEPT 2A

DESCRIPTIVE TITLE: CYLINDRICAL ABRASIVE SIEVE

SUPPORTING REQUIREMENTS:

DEPLOYMENT: BOOM DEPLOYED

SAMPLE TRANSPORT: PNEUMATIC - FLOW INDUCED IN TRANSPORT TUBE BY EJECTOR JET AT CYCLONE COLLECTOR

SAMPLE PROCESSING: DETERMINED BY SIZE OF OPENING IN ABRASIVE SIEVE.

OPERATIONAL CHARACTERISTICS:

POWER: SIMPLICITY:
EST. WGT: EST. VOL:

AFTER THE SAMPLER IS RESTING ON THE SURFACE THE MOTOR IS ACTIVATED TO ROTATE THE ABRASIVE SIEVE. A BRAKING FORCE AT THE HINGE POINT WILL CAUSE THE CYLINDER TO SLIP WITH RESPECT TO THE SURFACE WHILE ADVANCING THEREBY ABRADING THE SURFACE. SOIL PARTICLES ENTERING THE CYLINDER ARE CAPPED UP ON THE INNER WALL AND DROPPED INTO THE FLOW. THEY ARE THEN TRANSPORTED INTO THE CAPSULE THROUGH THE DEPLOYMENT BOOM. REVERSING THE MOTOR REVERSES THE DIRECTION OF TRAVEL.

QUALITY OF SAMPLE:

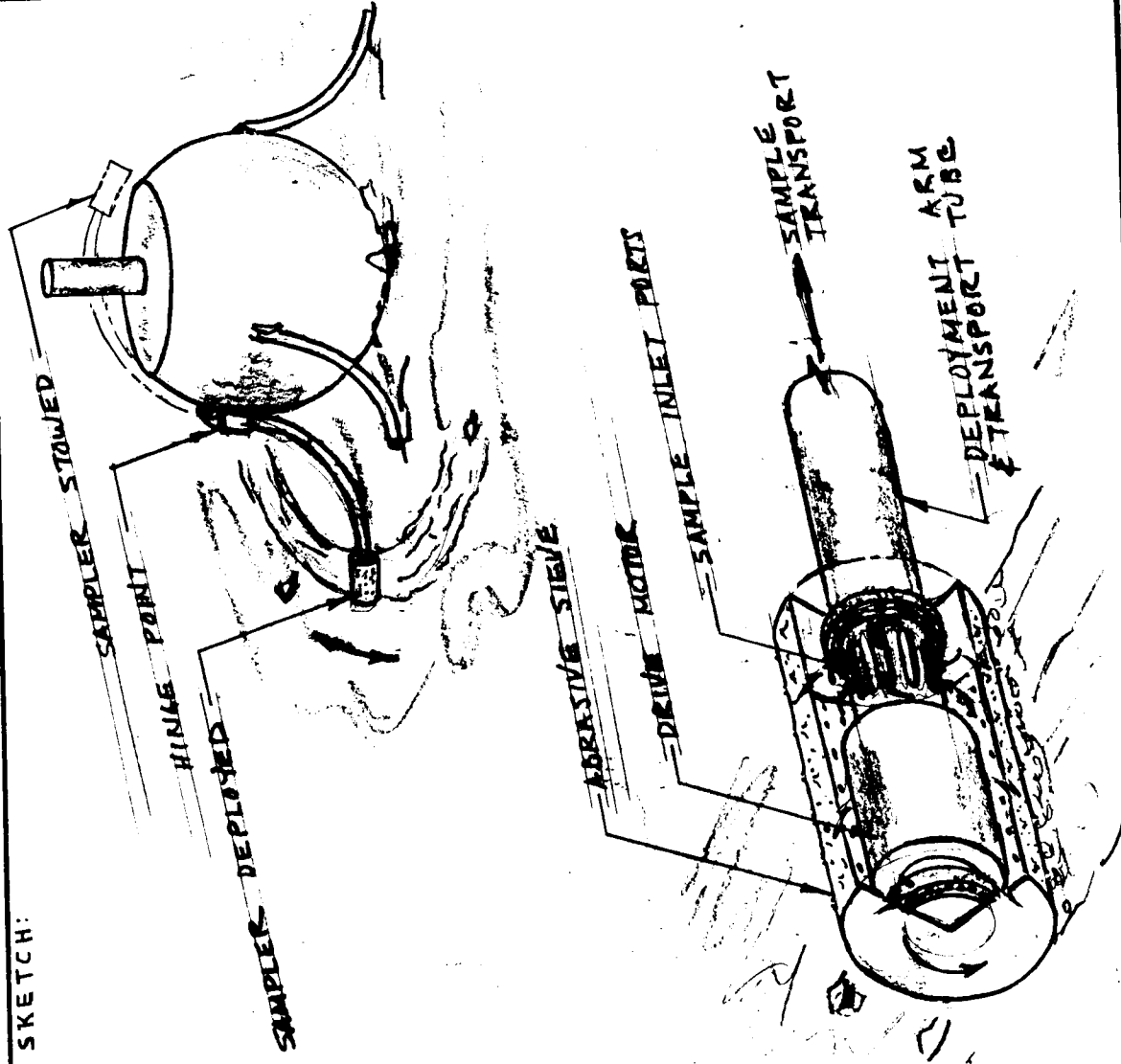
SAMPLE SIZE: 5-10 GRAMS
SAMPLE GRADE: DS 500μ
SAMPLE DIAS: WEAR OR FINE MATERIAL
SAMPLE DEGRADATION:

OVERALL PERFORMANCE:

SOIL MODELS: ALL
TERRAIN MODELS:
ENVIRONMENTAL:

REFERENCE DATA: JPL CONCEPT DATE: 3/16/69

SKETCH:



SAMPLING CONCEPT 2S

<p>DESCRIPTIVE TITLE: CONICAL ABRASIVE SIEVE</p>	<p>SKETCH:</p>
<p>SUPPORTING REQUIREMENTS: DEPLOYMENT: SHORT BOOM USING ROTATION AND VERTICAL EXTENSION. —</p> <p>SAMPLE TRANSPORT: MECHANICAL USING GRAVITY DROP</p>	
<p>SAMPLE PROCESSING: ACCOMPLISHED BY SIZE OF OPENINGS IN ABRASIVE PICKUP HEAD.</p> <p>OPERATIONAL CHARACTERISTICS: POWER: — EST. WGT: — EST. VOL: — SIMPLICITY: SIMPLE THE CONICAL ABRASIVE HEAD IS ROTATED WHILE THRUST IS BEING APPLIED. THE SOIL COLLECTS INSIDE THE CONICAL PICKUP HEAD. THE SOIL SAMPLE IS DELIVERED BY GRAVITY DROP BY ROTATING THE SAMPLER 180 DEGREES. ALLOWING THE SOIL TO FALL DOWN THE DEPLOYMENT TUBE.</p>	
<p>QUALITY OF SAMPLE: SAMPLE SIZE: 500gms SAMPLE GRADE: < 500µ SAMPLE DIAS: — SAMPLE DEGRADATION: —</p>	
<p>OVERALL PERFORMANCE: SOIL MODELS: ALL BUT SOLID ROCK TERRAIN MODELS: ANY ENVIRONMENTAL:</p>	
<p>REFERENCE DATA: DATE: 3/16/66</p>	

SAMPLING CONCEPT 26

DESCRIPTIVE TITLE: CONICAL ABRASIVE SIEVE		SKETCH: <div style="display: flex; justify-content: space-around; align-items: center;"> </div>
SUPPORTING REQUIREMENTS: DEPLOYMENT: VERTICAL		
SAMPLE TRANSPORT: MECHANICAL IN SIEVE AND BY CENTRIFUGAL FORCE		
SAMPLE PROCESSING: BY COLLECTION PROCESS-		
OPERATIONAL CHARACTERISTICS: POWER: SIMPLICITY: SIMPLE EST. WGT: EST. VOL:		
THE CONICAL ABRASIVE HEAD IS ROTATED WHILE THRUST IS BEING APPLIED. THE SOIL IS ABRASED BY OUTER SURFACE OF CONE. SMALL PERFCTIONS ALLOW SMALL PARTICLES TO COLLECT ON INNER SURFACE. THE SAMPLE IS TRANSFERRED TO CAPSULE VERTICALLY BY RETRACTING SAMPLER. FINAL TRANSFER IS ACCOMPLISHED BY SPINNING THE SAMPLER AT HIGH RPM TO SPIN OFF THE SAMPLE IN THE SHROUD WHICH COLLECTS THE SOIL IN THE SAMPLE CUP. THIS CUP IS STEPPED INTO POSITION AFTER THE SAMPLER HAS BEEN RETRACTED.		
QUALITY OF SAMPLE: SAMPLE SIZE: < 5 GRAM SAMPLE GRADE: 45504 SAMPLE DIAS. SAMPLE DEGRADATION:		
OVERALL PERFORMANCE: SOIL MODELS: ALL BUT SOLID ROCK TERRAIN MODELS: ENVIRONMENTAL:		
REFERENCE DATA:		DATE: 3/17/66

SAMPLING CONCEPT 27

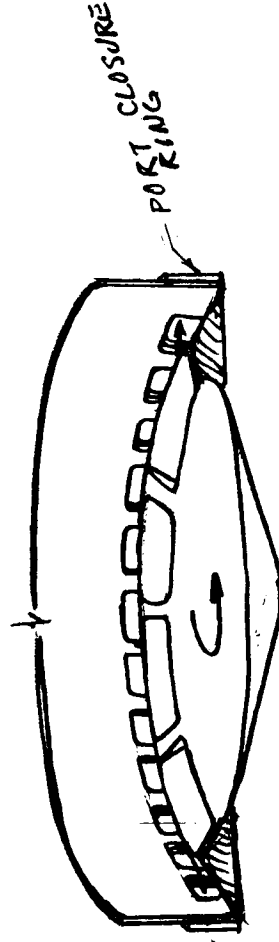
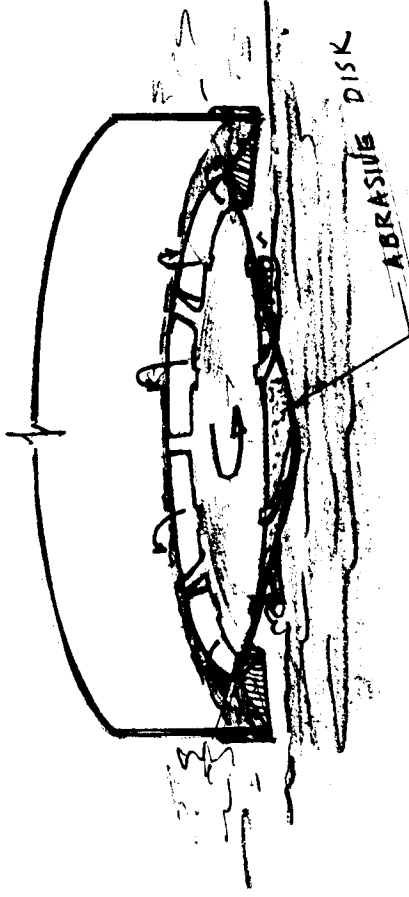
DESCRIPTIVE TITLE: SPHERICAL ABRADING SIEVE (TYPE A)	<p>SKETCH:</p>
SUPPORTING REQUIREMENTS: DEPLOYMENT: FURLABLE BOOM. DEPLOYED LENGTH 10-20 FEET	
SAMPLE TRANSPORT: PNEUMATIC OR GRAVITY DROP	
SAMPLE PROCESSING: BY ACTION OF COLLECTION MECHANISM	
OPERATIONAL CHARACTERISTICS: POWER: 25-50 WATTS EST. WGT: THE SPHERICAL ABRADING HEAD IS DEPLOYED VERTICALLY TO FULL LENGTH OF BOOM. BOOM IS THEN ROTATED UNTIL SAMPLING HEAD CONTACTS SURFACE. THE SAMPLING HEAD IS THEN ROTATED AS BOOM IS SLOWLY RETRACTED TO COLLECT SAMPLE. THE SAMPLING HEAD IS THUS FREE TO FOLLOW THE LOCAL TERRAIN CONTOURS WHILE SAMPLING. SOIL TRANSPORT IS BY PNEUMATIC FLOW INSIDE FURLABLE BOOM. AN ALTERNATE METHOD IS TO ALLOW SOIL TO COLLECT IN SPHERICAL HEAD AND THEN ROTATING THE BOOM TO A VERTICAL POSITION TO TRANSPORT THE SOIL SAMPLE BY GRAVITY DROP INSIDE BOOM TUBE.	
QUALITY OF SAMPLE: SAMPLE SIZE: GRAMS SAMPLE DIAS: SAMPLE DEGRADATION:	
OVERALL PERFORMANCE: SOIL MODELS: ALL MODELS TERRAIN MODELS: ALL MODELS ENVIRONMENTAL:	
REFERENCE DATA: JPL LAB & AERONAUTRONIC DATE: 2/25/66	

SAMPLING CONCEPT 28

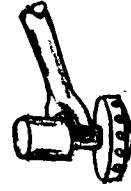
<p>DESCRIPTIVE TITLE: ABRASIVE DISK WITH CENTRIFUGAL COLLECTION</p>	<p>SUPPORTING REQUIREMENTS: DEPLOYMENT: VERTICALLY OR HORIZONTALLY ON A BOOM</p>	<p>SAMPLE TRANSPORT: MECHANICAL BY CENTRIFUGAL ACTION</p>	<p>SAMPLE PROCESSING: ABRASION</p>	<p>OPERATIONAL CHARACTERISTICS:</p>	<p>POWER: EST WGT: SIMPLICITY: SIMPLE EST. VOL:</p>	<p>THE ABRADING DISK FIXED TO THE END OF A TUBE IS FORCED INTO THE SURFACE AS IT ROTATES. THE SOIL SAMPLE IS COLLECTED AT THE OUTER EDGE OF THE DISK BY CENTRIFUGAL ACTION. TO TRANSFER THE SAMPLE THE SAMPLE PORT CLOSURE RING IS ROTATED TO OPEN THE PERIPHERAL PORTS AND CENTRIFUGAL ACTION IS USED TO TRANSPORT THE SAMPLE.</p>	<p>QUALITY OF SAMPLE: SAMPLE SIZE: GRAMS SAMPLE DIAS: FINES SAMPLE DEGRADATION: THERMAL</p>	<p>OVERALL PERFORMANCE: SOIL MODELS: ALL BUT SOLID ROCK TERRAIN MODELS: ENVIRONMENTAL:</p>	<p>REFERENCE DATA: DATE: 3/14/66</p>
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SKETCH:

SAMPLE COLLECTION



SAMPLE DUMP



BOOM MOUNTED

SAMPLING CONCEPT 29

DESCRIPTIVE TITLE: INFLATABLE SLOTTED TUBE

SUPPORTING REQUIREMENTS:

DEPLOYMENT: TUBING CAN BE DEPLOYED BALLISTICALLY OR SIMPLY DROPPED TO THE SURFACE

SAMPLE TRANSPORT: MECHANICALLY BY REELING IN THE TUBE. PASSING THE TUBE THROUGH PINCH ROLLS TO OPEN THE SLOTS RELEASES THE SOIL FOR TRANSFER TO THE CAPSULE

SAMPLE PROCESSING: PARTICLE SIZE DETERMINED BY SIZE OF OPENING

OPERATIONAL CHARACTERISTICS:

POWER: NIL
EST. WGT: SIMPLICITY: SIMPLE
EST. VOL:

THE SAMPLER CONSISTS OF TWO CONCENTRIC ELASTOMERIC TUBES. THE OUTER TUBE IS SLOTTED AT INTERVALS ALONG ITS LENGTH. INFLATING THE INNER TUBE CAUSES IT TO BULGE OUT AT THE SLOTS CAUSING THE SLOTS TO WIDEN OR OPEN. SOIL PARTICLES CAN THEN FALL INTO THE OPEN SLOTS. THIS CAN BE ENCOURAGED BY PULSING THE PRESSURE IN THE INNER TUBE AND BY DRAGGING THE TUBING ALONG THE SURFACE. REMOVING THE PRESSURE IN THE INNER TUBE ALLOWS THE SLOTS TO CLOSE TRAPPING THE SOIL PARTICLES. THE TUBE IS UNLOADED BY PASSING IT THROUGH PINCH ROLLS

QUALITY OF SAMPLE:

SAMPLE SIZE: < 5 GRAMS
SAMPLE GRADE: AS 2MM
SAMPLE DIAS: WILL TEND TO LOSE THE FINES.
SAMPLE DEGRADATION: NONE

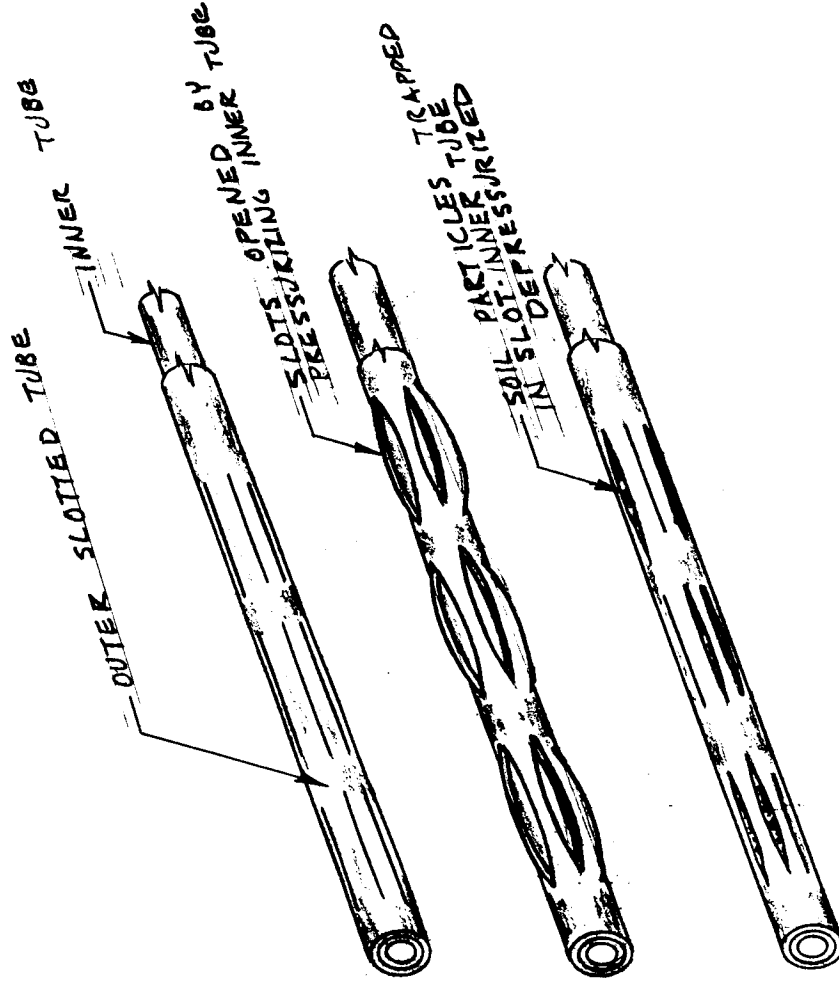
OVERALL PERFORMANCE:

SOIL MODELS: LOOSE AGGREGATE
TERRAIN MODELS:
ENVIRONMENTAL:

REFERENCE DATA: JPL CONCEPT

DATE: 3/18/66

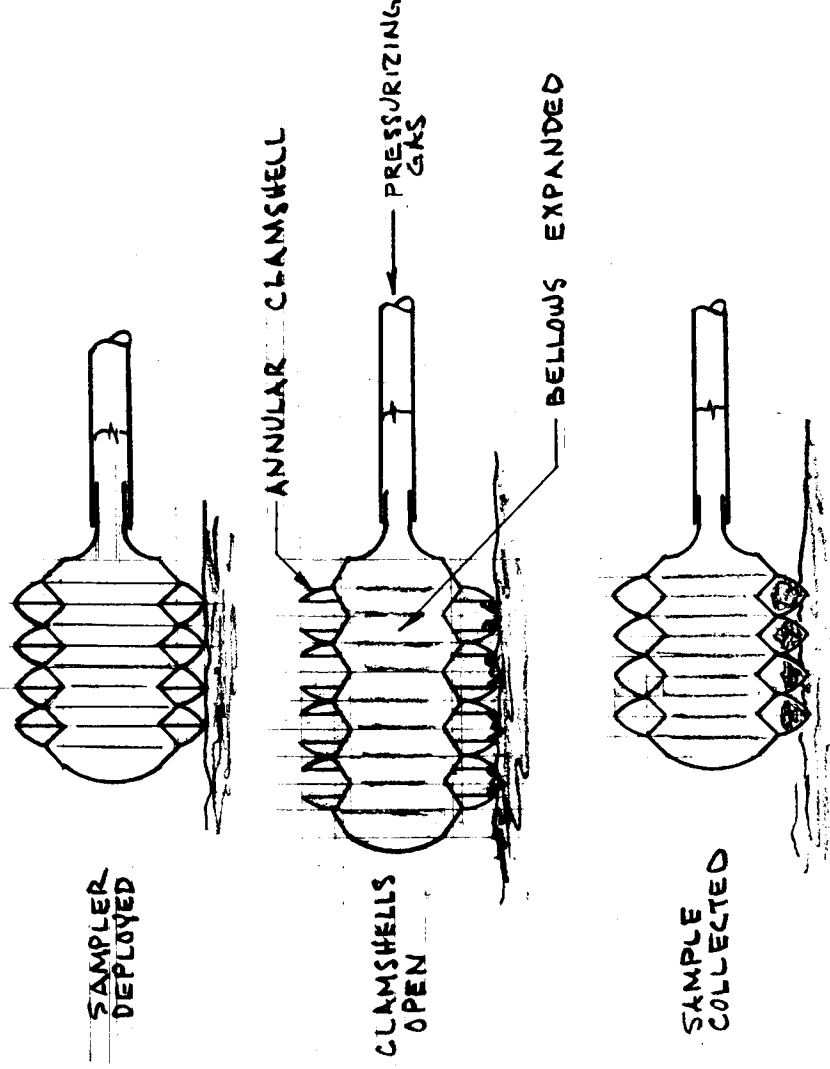
SKETCH:



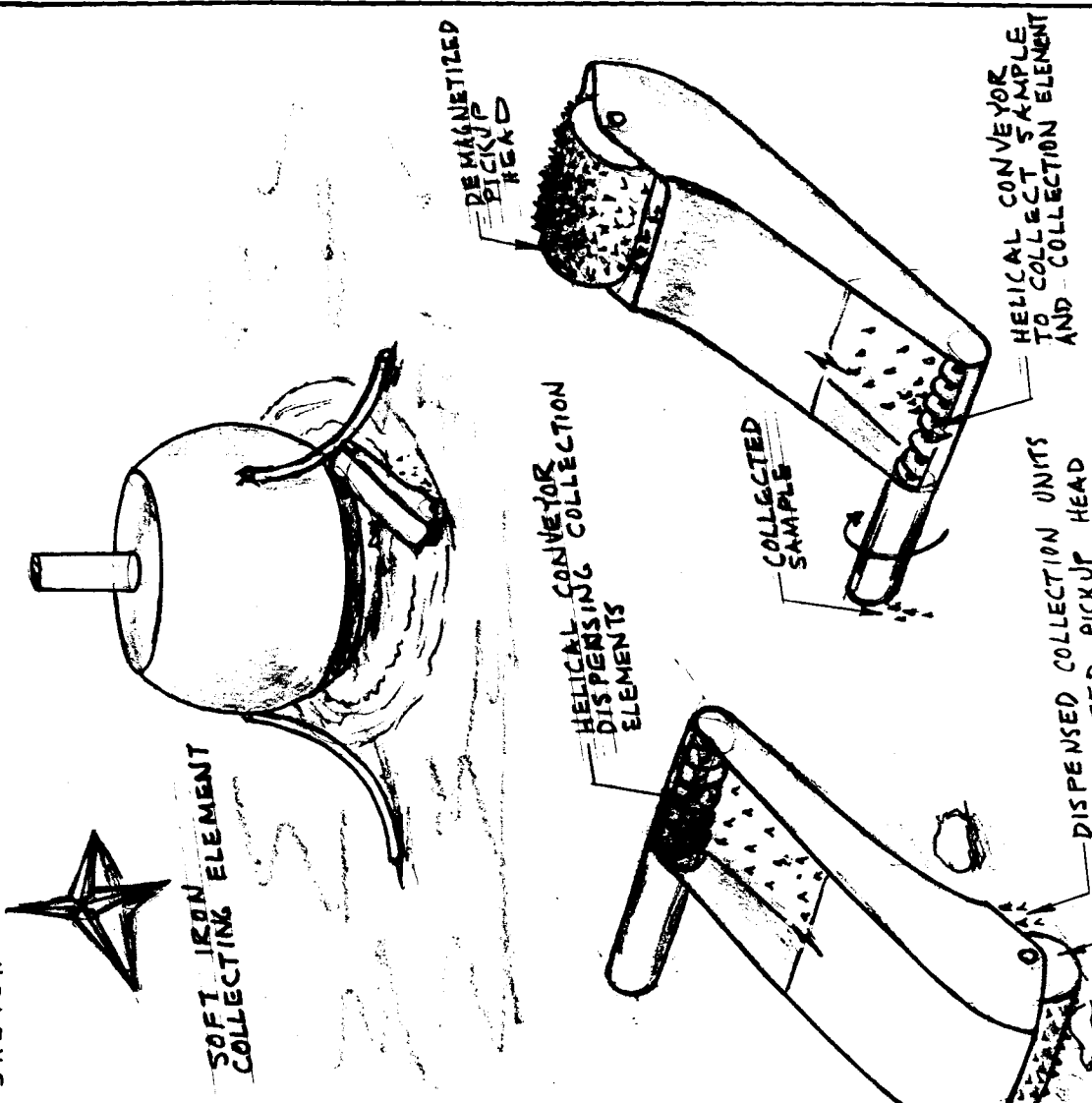
SAMPLING CONCEPT 30

<p>DESCRIPTIVE TITLE: BELLOWS ACTUATED CLAMSHELL</p>	
<p>SUPPORTING REQUIREMENTS: DEPLOYMENT: BALLISTIC OR CONVEYOR BELT.</p>	
<p>SAMPLE TRANSPORT: MECHANICAL</p>	
<p>SAMPLE PROCESSING: SIZE LIMITED BY DIMENSIONS OF CLAMSHELLS</p>	
<p>OPERATIONAL CHARACTERISTICS:</p>	
<p>POWER: NIL</p>	<p>SIMPLICITY: SIMPLE</p>
<p>EST WGT: NIL</p>	<p>EST VOL: NIL</p>
<p>ANNULAR CLAMSHELL RINGS ARE ATTACHED TO THE CRESSES OF A METAL BELLOWS. THE CLOSED CLAMSHELLS CAN BE DRAWN ALONG THE SURFACE AND OPENED AT WILL BY PRESSURIZING THE BELLOWS. MULTIPLE UNITS CAN BE CONNECTED TO FORM A STRUNG OR ENDLESS CONVEYOR. RELEASING THE PRESSURE ALLOWS THE BELLOWS TO COLLAPSE TRAPPING SOIL IN THE CLAMSHELLS.</p>	
<p>QUALITY OF SAMPLE:</p>	
<p>SAMPLE SIZE: SMALL</p>	<p>SAMPLE GRADE:</p>
<p>SAMPLE DIAS: NONE</p>	<p>SAMPLE DEGRADATION: NONE</p>
<p>OVERALL PERFORMANCE:</p>	
<p>SOIL MODELS: ALL BUT SOLID ROCK</p>	<p>TERRAIN MODELS: ALL ENVIRONMENTAL</p>
<p>REFERENCE DATA:</p>	<p>DATE: 3/14/66</p>

SKETCH:



SAMPLING CONCEPT 31

<p>DESCRIPTIVE TITLE: MAGNETIC SAMPLER</p>	<p>SUPPORTING REQUIREMENTS: DEPLOYMENT: SHORT BOOM OR SKAF</p> <p>SAMPLE TRANSPORT: MECHANICALLY BY ENTRAPMENT WITH SOFT IRON COLLECTION ELEMENT ABOUT .06 INCHES PER SIDE</p> <p>SAMPLE PROCESSING:</p>	<p>SKETCH:</p> 
<p>OPERATIONAL CHARACTERISTICS:</p> <p>POWER: SIMPLICITY: MODERATE</p> <p>EST WGT: EST VOL:</p> <p>SOFT IRON COLLECTION ELEMENTS SHAPED LIKE TETRAHEDRONS ARE DISTRIBUTED IN FRONT OF A ROLLING ELECTROMAGNETIC PICKUP HEAD. AS THE HEAD ROLLS ALONG THE SURFACE IT PICKS UP THE SOFT IRON ELEMENTS WITH ADHERING SOIL. AS IT CONTINUES TO ROLL ADDITIONAL SOIL IS IMPACTED BETWEEN THE COLLECTED ELEMENT TO DUMP THE SAMPLE. THE SAMPLING HEAD AND SHROUD IS ROTATED 180° AND THE PICKUP HEAD DEMAGNETIZED TO RELEASE THE SOFT IRON COLLECTION ELEMENTS AND THE ENTRAPPED SOIL. THE SHROUD ACTS AS A DELIVER CHUTE TO THE HELICAL CONVEYOR.</p>	<p>QUALITY OF SAMPLE:</p> <p>SAMPLE SIZE: SMALL SAMPLE GRADE:</p> <p>SAMPLE DIAS: MAGNETIC MATERIAL CONCENTRATED</p> <p>SAMPLE DEGRADATION:</p>	<p>OVERALL PERFORMANCE:</p> <p>SOIL MODELS:</p> <p>TERRAIN MODELS:</p> <p>ENVIRONMENTAL:</p> <p>REFERENCE DATA:</p> <p>DATE: 2/25/66</p>

SAMPLING CONCEPT 32

DESCRIPTIVE TITLE: WIRE BRUSH CONVEYOR

SUPPORTING REQUIREMENTS:

DEPLOYMENT: TELESCOPING BOOM - RANGE
5-10 FEET

SAMPLE TRANSPORT: MECHANICAL PLUS
PNEUMATIC

SAMPLE PROCESSING: PARTLY BY ACTION
OF BRUSH CONVEYOR AND PARTLY BY
PNEUMATIC SEPARATION OF SAMPLE FROM
CONVEYOR

OPERATIONAL CHARACTERISTICS:

POWER:

EST WGT:

SIMPLICITY:

EST. VOL:

AFTER EXTENDING THE TELESCOPING CONVEYOR
TUBE, IT IS ROTATED UNTIL THE END
RESTS ON THE SURFACE. THIS DEPLOYS
A CHAIN LINK WIRE BRUSH CONVEYOR ON THE
SURFACE. ACTIVATING A CAPSTAN DRIVE WILL
PULL THE CONVEYOR ALONG THE SURFACE
CARRYING SOIL TOWARDS THE CONVEYOR
TUBE INLET. AS SOIL COLLECTS AT THE
INLET, PORTIONS ARE CARRIED ALONG
BETWEEN BRUSHES UNTIL IT REACHES THE
PNEUMATIC TRANSFER HEAD WHERE A
HIGH VELOCITY JET BLOWS THE SOIL OUT
OF THE CONVEYOR INTO A CYCLONE
COLLECTOR.

QUALITY OF SAMPLE:

SAMPLE SIZE: GRAMS SAMPLE GRADE:

SAMPLE DIAS:

SAMPLE DEGRADATION:

OVERALL PERFORMANCE:

SOIL MODELS: ALL MODELS

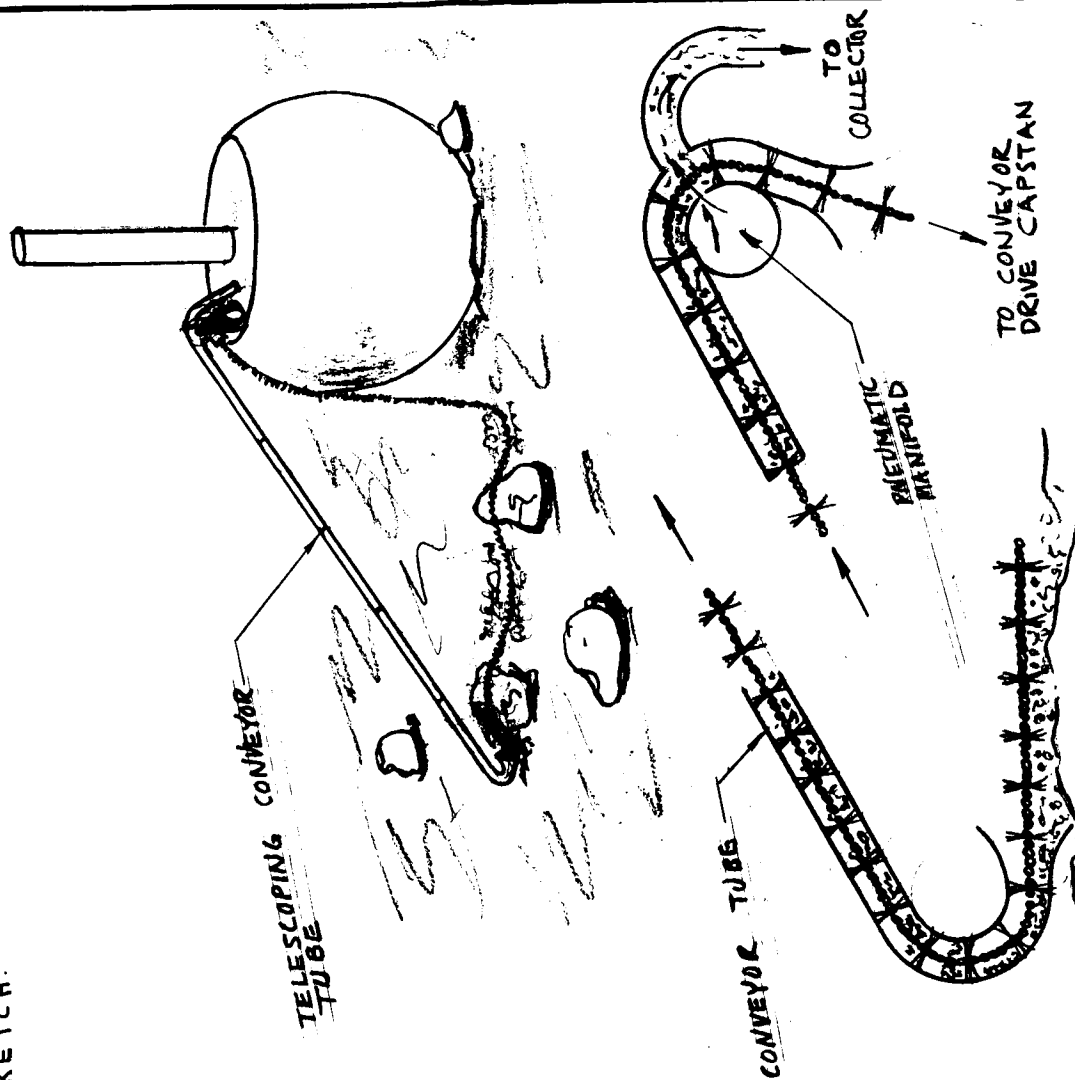
TERRAIN MODELS: ALL MODELS

ENVIRONMENTAL:

REFERENCE DATA:

DATE: 3/8/66

SKETCH:



APPENDIX C

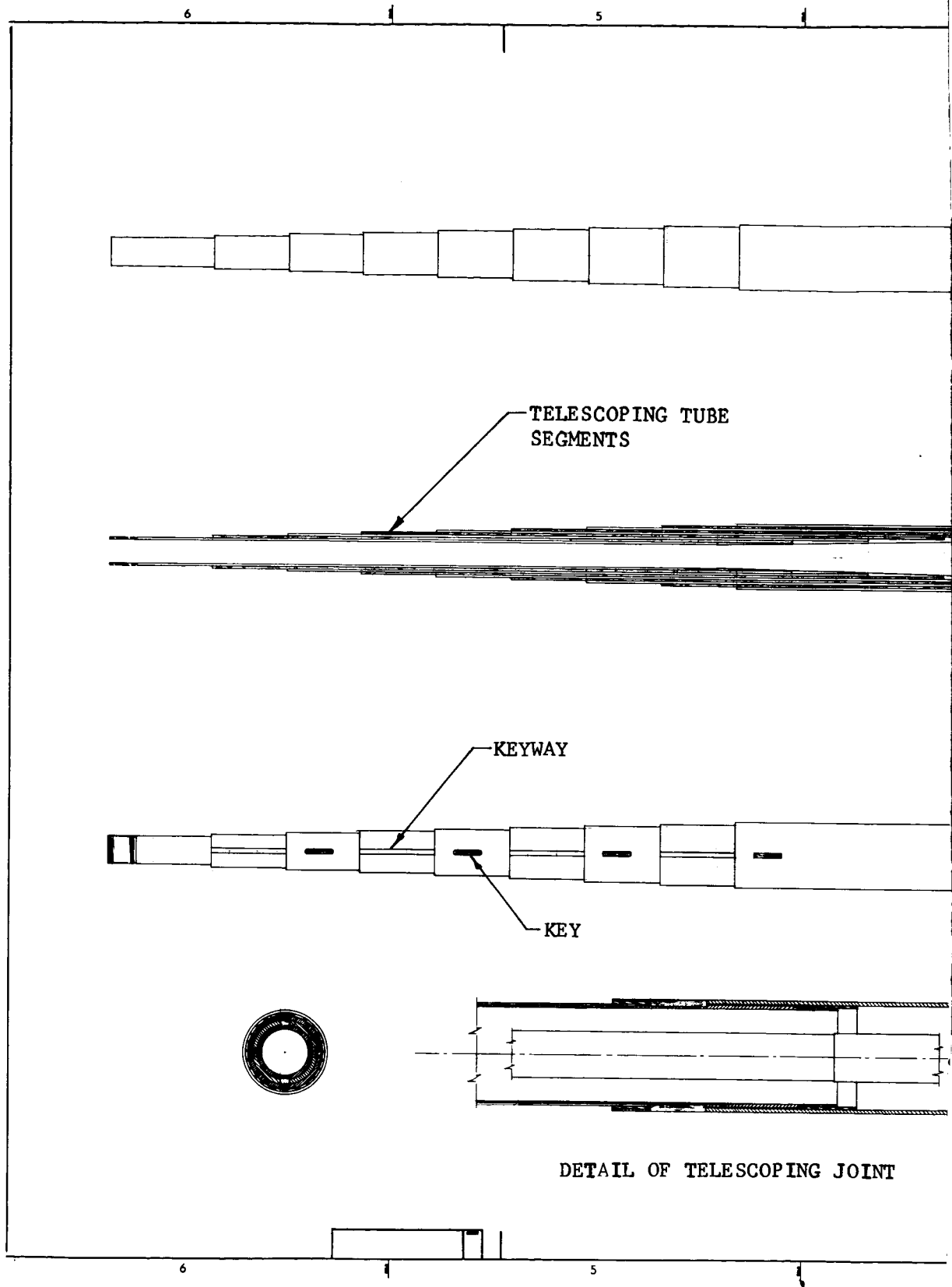
PRELIMINARY DESIGNS

This appendix contains the eight general concepts which were selected for more detailed preliminary design effort to develop the mechanization of the design. These drawings served as the basis for a more detailed second order evaluation from which the most promising of four concepts were selected for further design development. These four concepts are identified as follows:

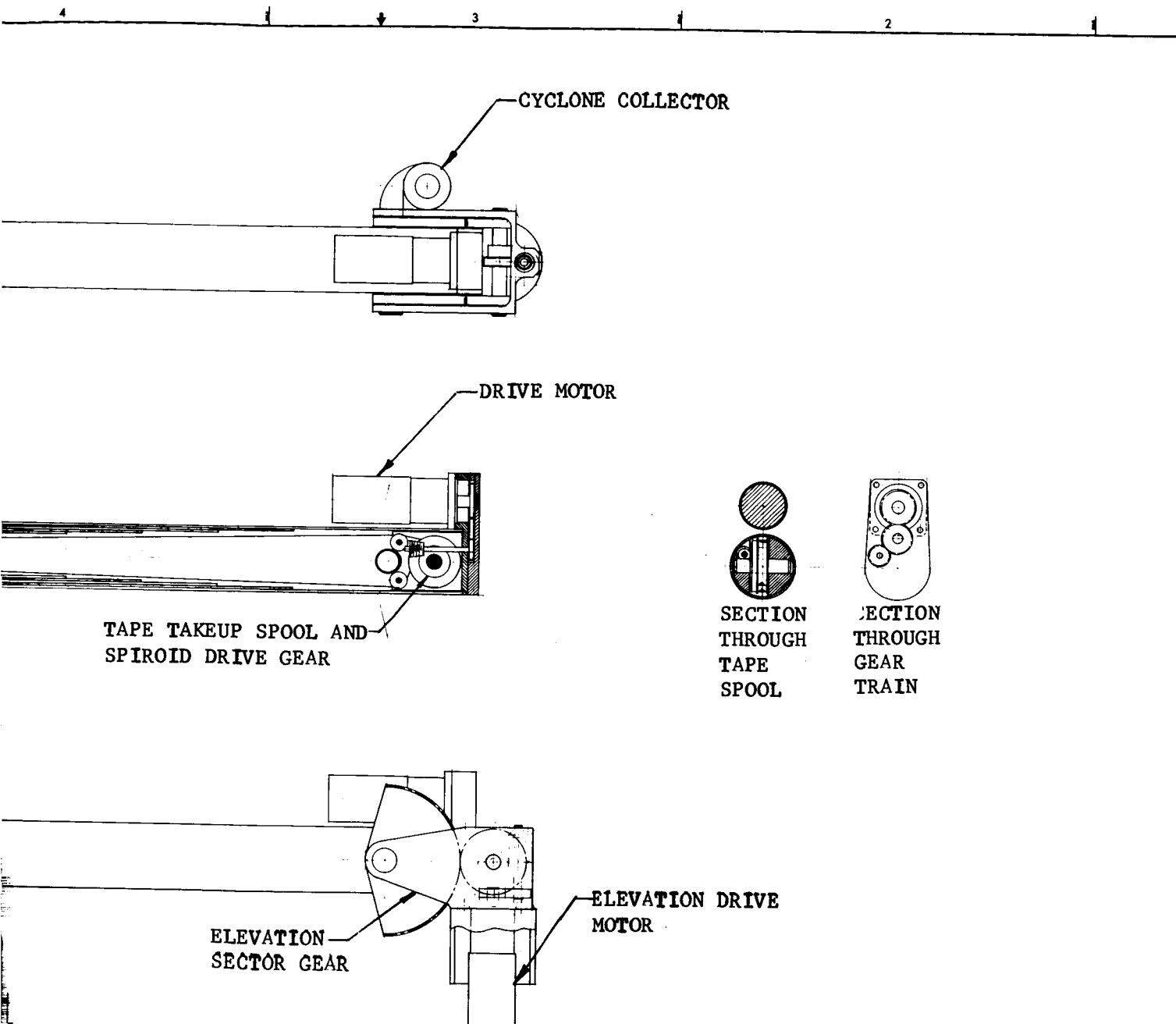
- 5D Rotary Scoop PD25167
- 7A Rotating Wire Brush PD25174 and PD25199
- 8B2 Conical Abrasive Sieve PD25166
- 8C Spherical Abrading Head PD25173

The telescoping boom (PD33918) was selected as the deployment mechanism for concepts 7A and 8C. The furlable boom was felt to involve too much development.

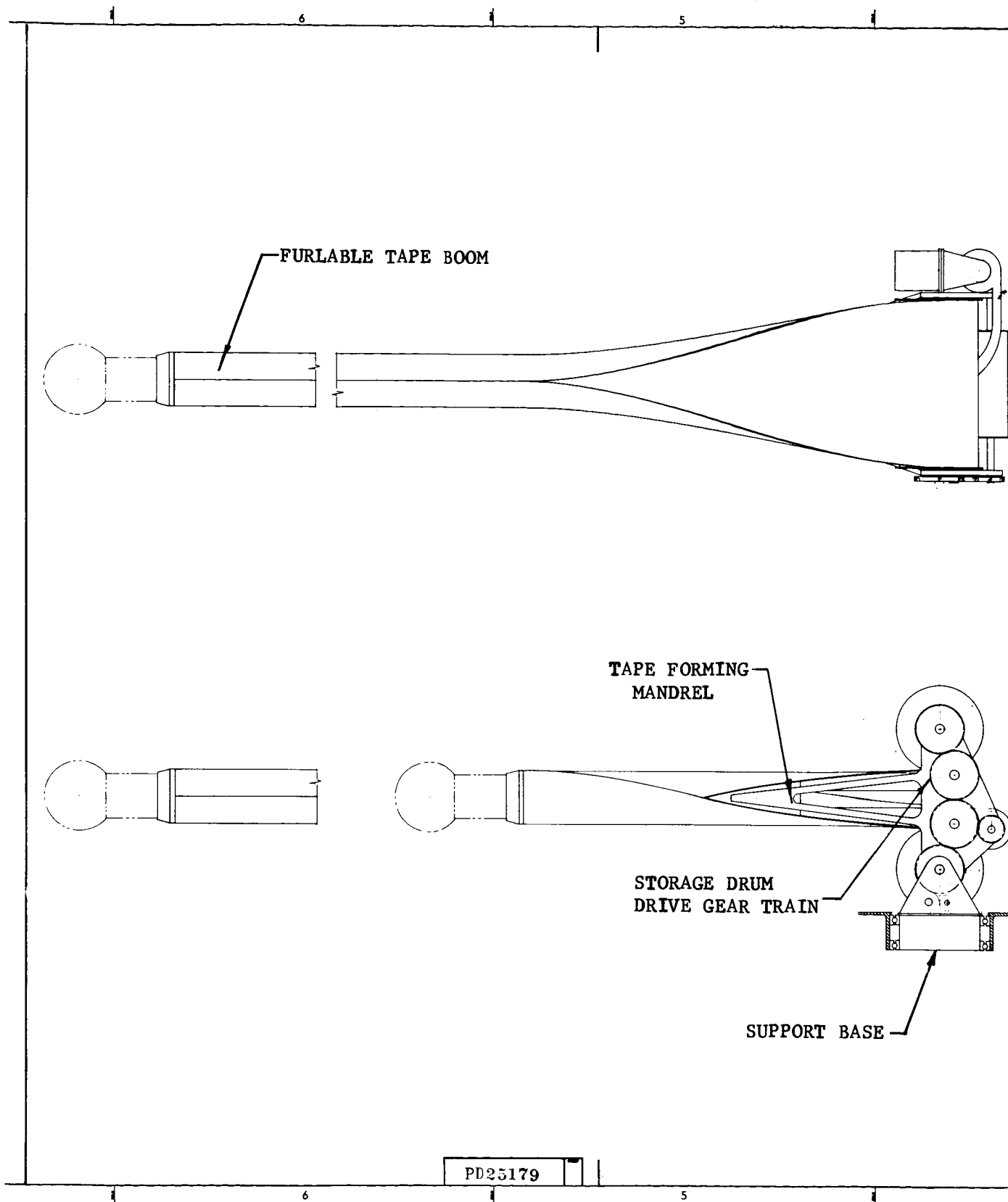
It quickly became apparent that the schedule was so short that the development of all these concepts could not be carried on simultaneously. Another evaluation then reduced the soil sampler development to one vertically deployed sampler as typified by concept 8B2, the conical abrasive sieve, and one horizontally deployed sampler as typified by concept 7A, the rotating wire brush mounted on the end of the telescoping boom.



C-2-1



QTY REQD	IDENTIFYING NUMBER	REFERENCE NUMBER	NOMENCLATURE	DESCRIPTION	SIZE	SPECIFICATION	ZONE	ITEM NO.
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DIMENSIONS PER MIL-STD-8			CHECK					
DIMENSIONS ARE IN INCHES			ENGR					
TOLERANCES ON			STRESS					
DECIMALS			WEIGHT					
1/16 = .0625			PROJ					
1/32 = .03125			APPD					
1/64 = .015625			APPROVED					
SURFACE TEXTURE			APPROVED					
PER ASA B46.1-1962			APPROVED					
DO NOT SCALE THIS DRAWING			APPROVED					
RELEASE DATE			APPROVED					
PART NUMBER	NEXT ASSY	USED ON	NEXT FINAL ASSY	CODE IDENT NO				
APPLICATION	QTY REQD			09205 J				
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				PE33918				
				SCALE 1/1				
				WEIGHT				
				SHEET				



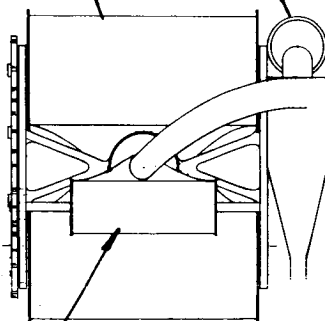
C-3-1

4 3

CYCLONE COLLECTOR

BOOM STRUCTURE
FURLABLE TAPES

PNEUMATIC TRANSPORT
TUBE FURLABLE TAPE



PD25179

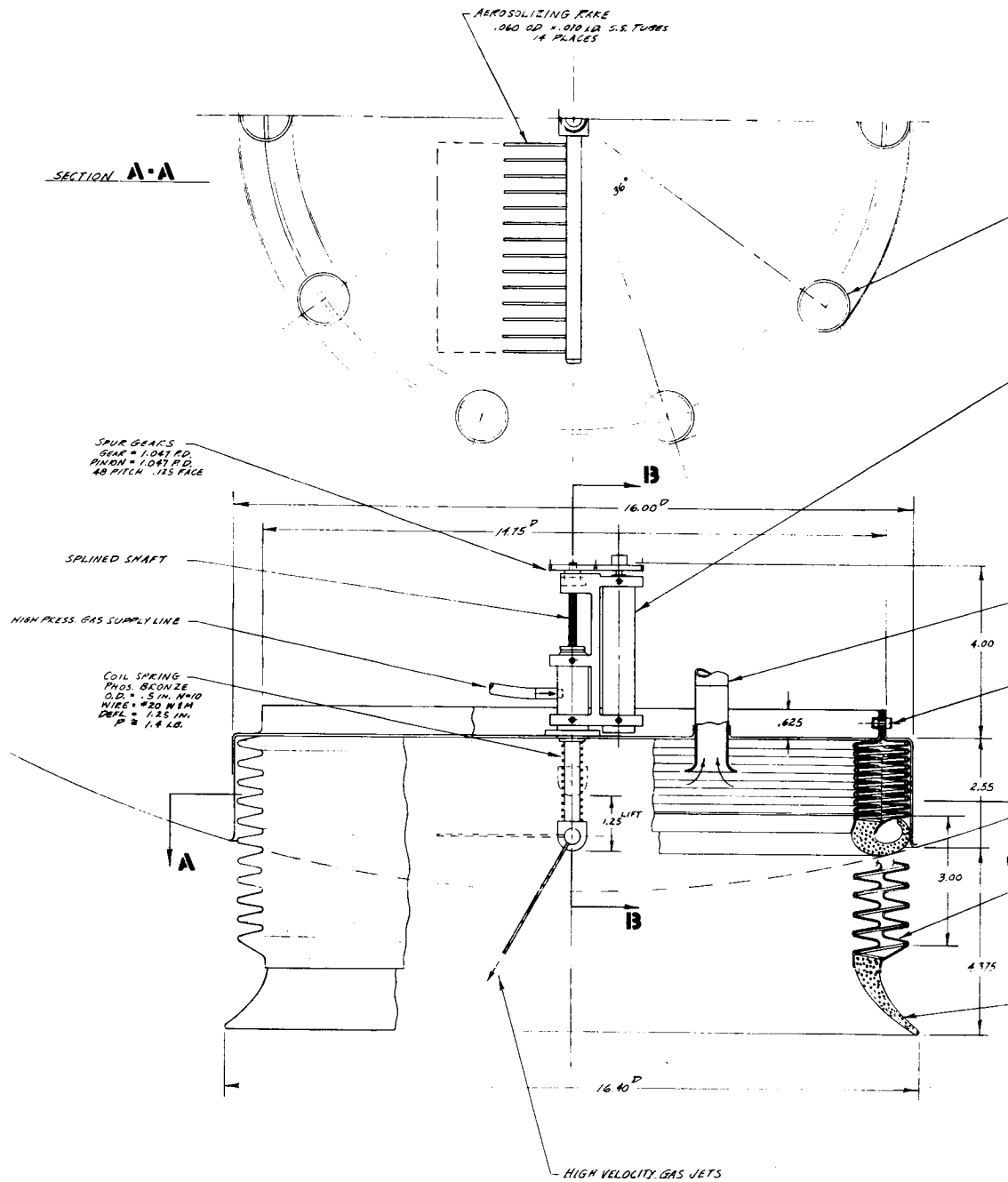
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DIMENSIONS ARE IN INCHES				CHECK					
TOLERANCES ON				ENGR					
DECIMALS				STRESS					
X = .05				WEIGHT					
XX = .03				PROJ					
XXX = .010				APPR					
SURFACE									
ROUGHNESS									
PER MIL STD-10									
DO NOT SCALE THIS DRAWING				APPROVED					
RELEASE DATE				APPROVED					
PART NUMBER				CODE IDENT NO		SIZE		PD25179	
NEXT ASSY				09205		J			
USED ON				SCALE		WEIGHT		SHEET	
APPLICATION									

St. Louis Engineering
AIRCRAFT/NAVY/SPACE DIVISION
NEW ORLEANS, LOUISIANA

FURLABLE BOOM
CONCEPT

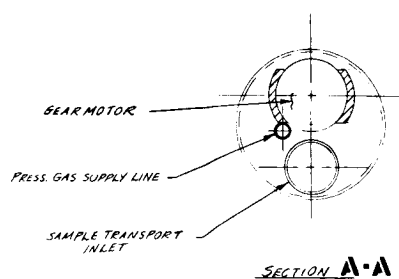
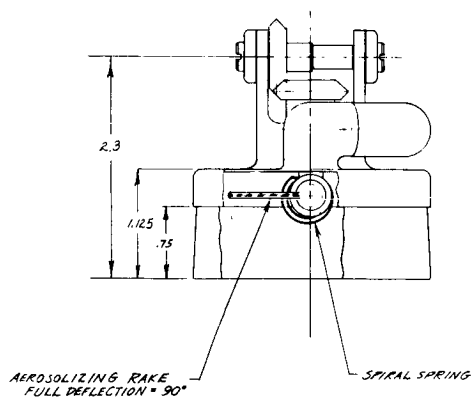
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SECTION A-A



C-4-1

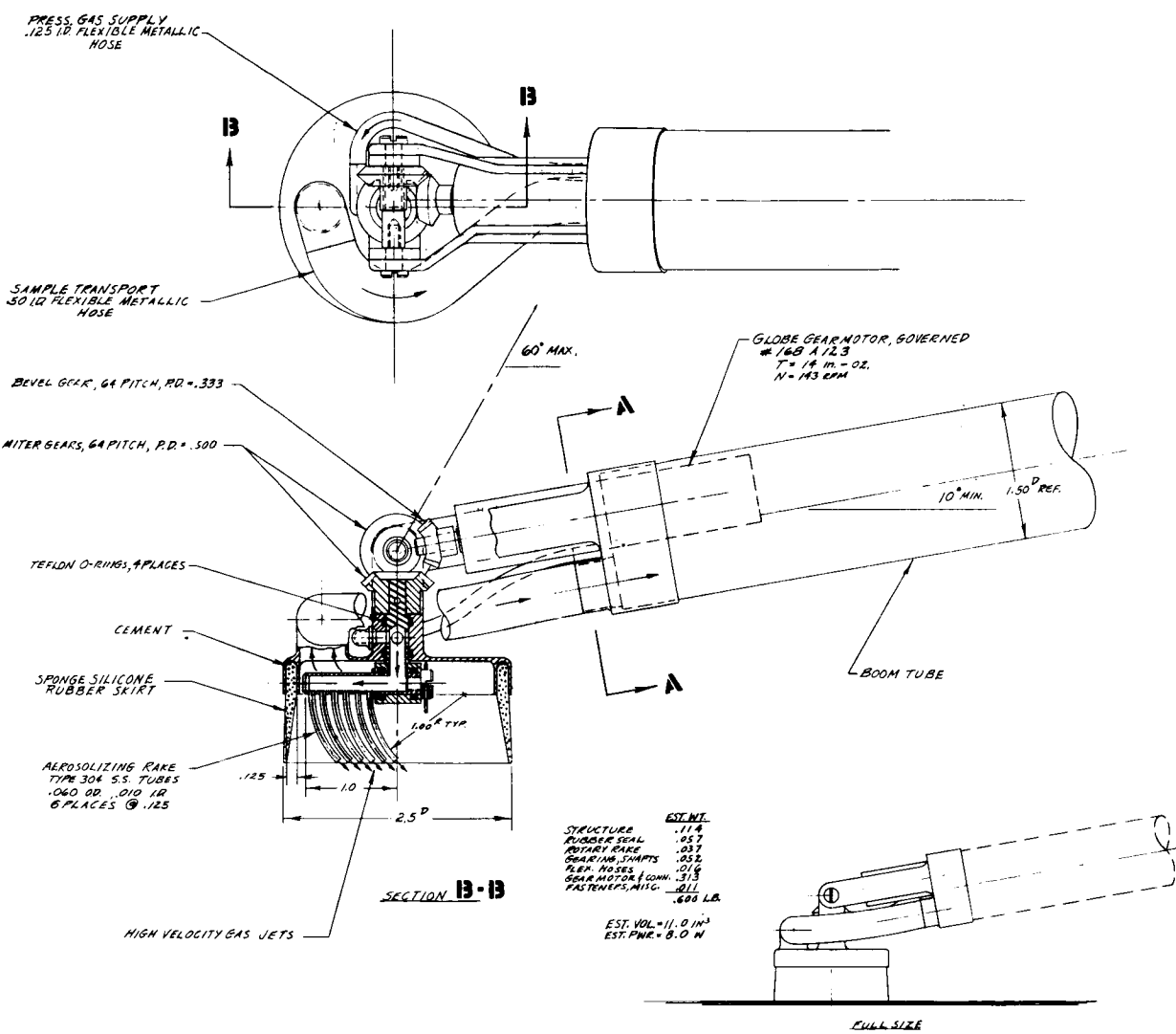
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10-70-7010

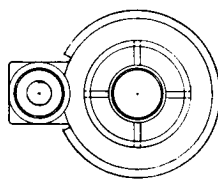
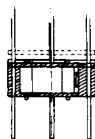
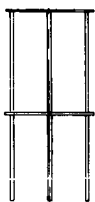
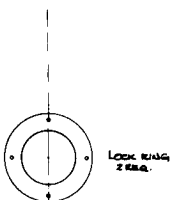
C-5-1

REVISIONS			
REV.	DESCRIPTION	DATE	APPROVED

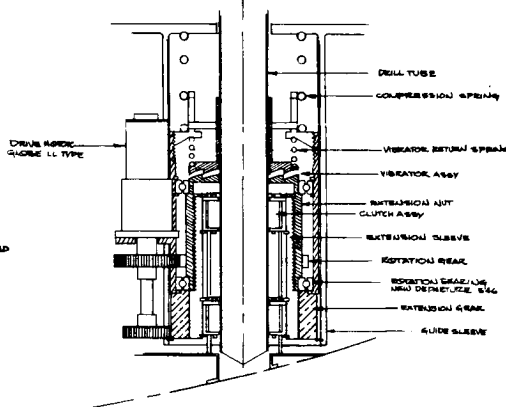


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TOLERANCES ON		ENG'R	SAMPLING CONCEPT 2A-2	
DECIMALS	ANGLES	STRESS	TYPE A	
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1/32" = .03125"	1/32° = .75°	PROL	SIZE E	
1/64" = .015625"	1/64° = .375°	APPR	PD25175	
SURFACE ROUGHNESS PER MIL-STD-10		APPROVED	SCALE 2" X WEIGHT	
DO NOT SCALE THIS DRAWING		APPROVED	SHEET	
PART NUMBER	NEXT ASSY	USED ON	QTY REQD	DATE

REVISIONS			
REV	DESCRIPTION	DATE	APPROVED



STOWED POSITION

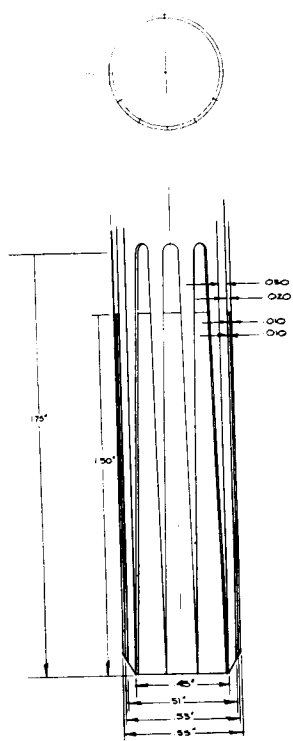


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		ANGLES		
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		XX = .03		
		XXX = .02		
		SURFACE		
		ROUGHNESS		
		PER MIL/STO 10		
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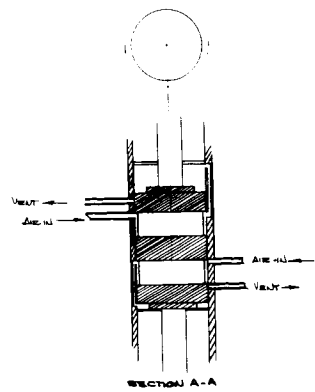
22 WIND. IFA 90032 V.001 PD25167

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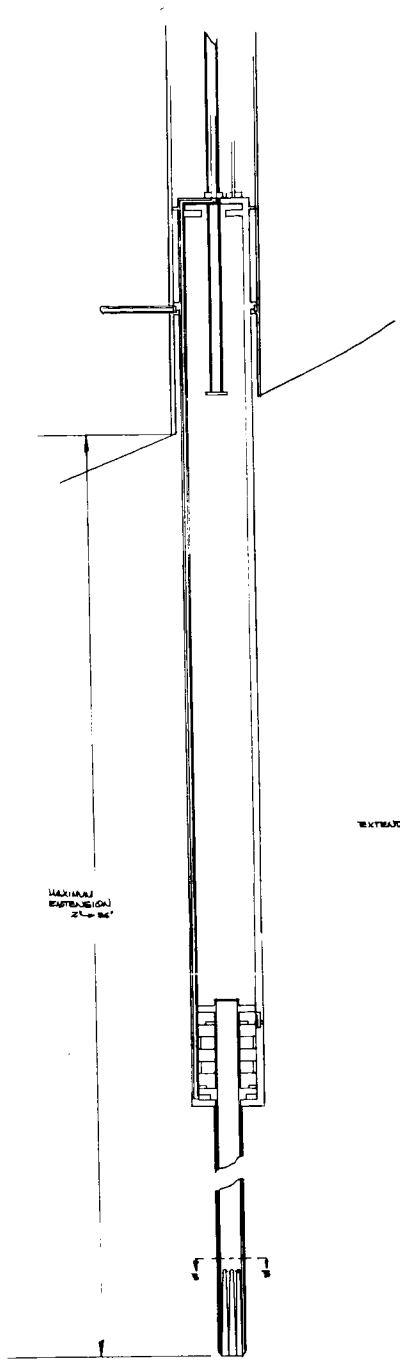
D
C
B
A



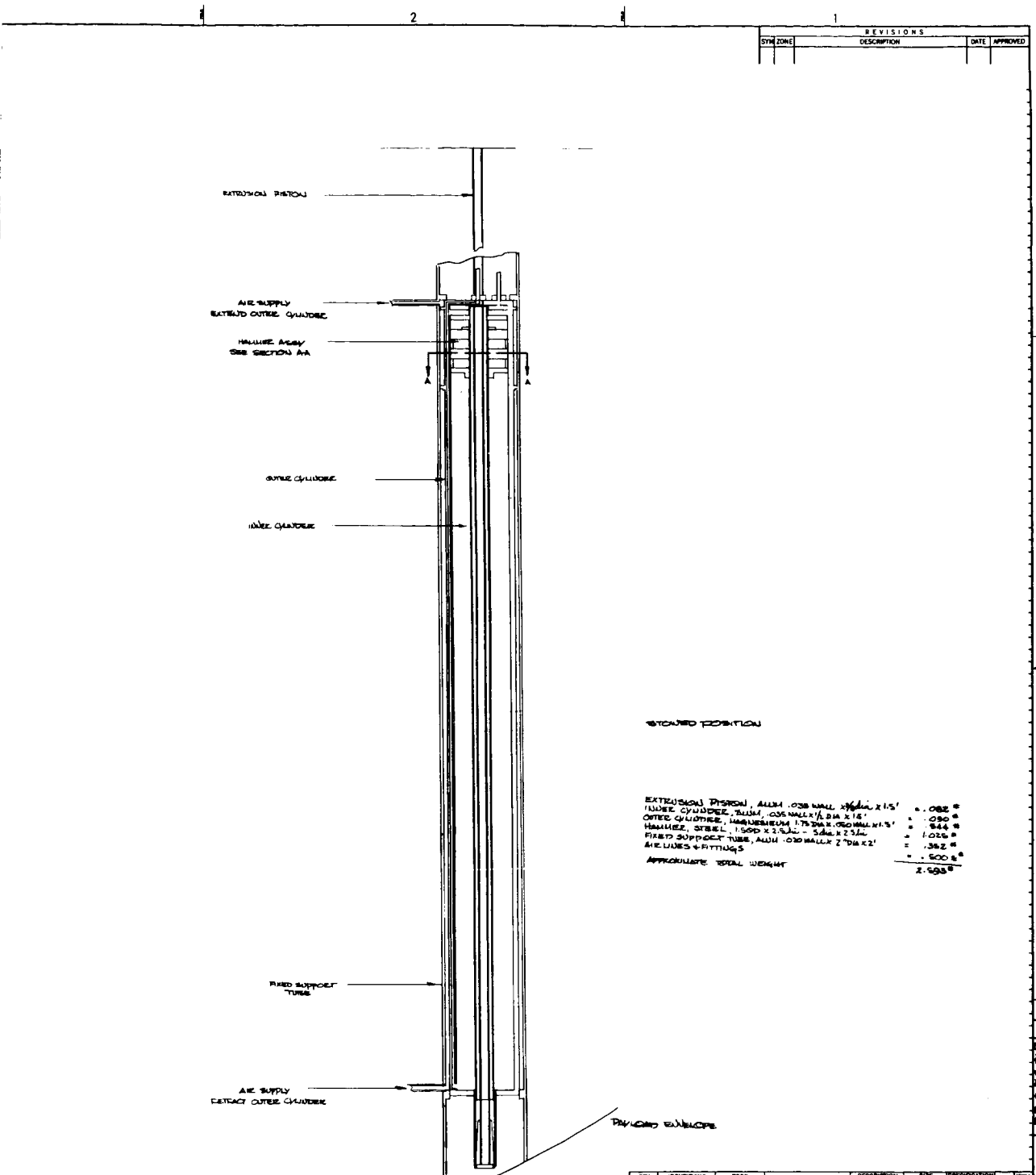
SECTION B-B
SCALE X 5



SECTION A-A



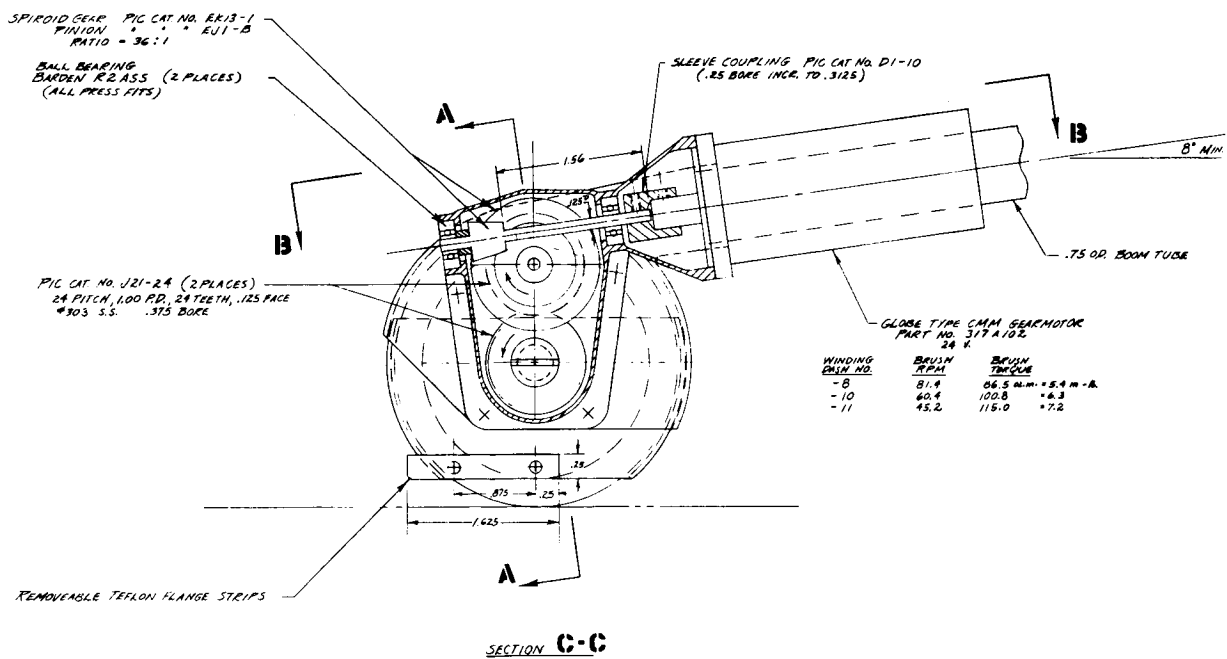
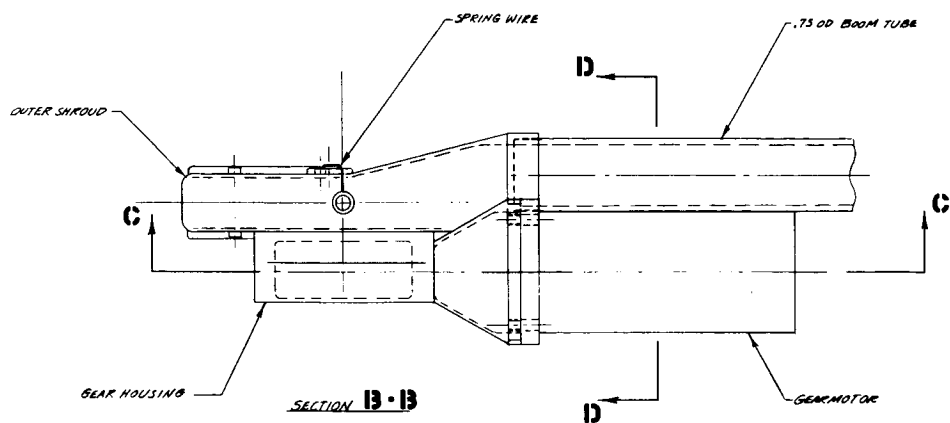
C-7-1



EXTRUSION PISTON, ALUM. .030 WALL 2 7/8" X 15'	• .082 #
INNER CYLINDER, TITANIUM, .030 WALL 1 1/2" DIA X 15'	• .080 #
OUTER CYLINDER, INCONEL 1.75 DIA X .050 WALL X 15'	• .944 #
HAIRLINE, STEEL, 1.500 X 2 1/2" DIA - 3 DIA X 2 1/2"	• 1.016 #
FIXED SUPPORT TUBE, ALUM. .030 WALL 2 7/8" X 2'	• .382 #
FITTINGS & FITTINGS	• .500 #
APPROXIMATE TOTAL WEIGHT	2.998 #

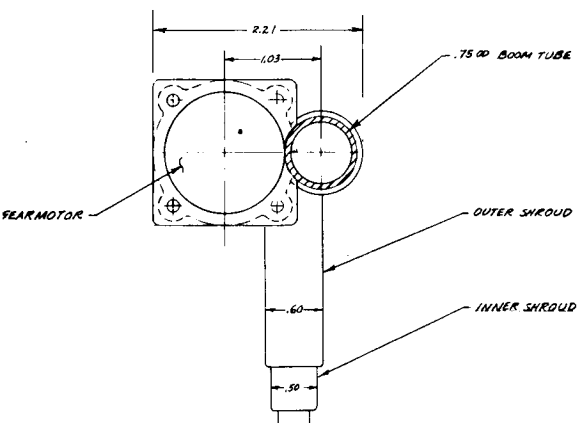
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SURFACE ROUGHNESS PER MIL-STD-10				APPROVED 09205 J PD25176 SCALE WEIGHT SHEET			
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APPLICATION			QTY REQD	RELEASE DATE			

PD25176

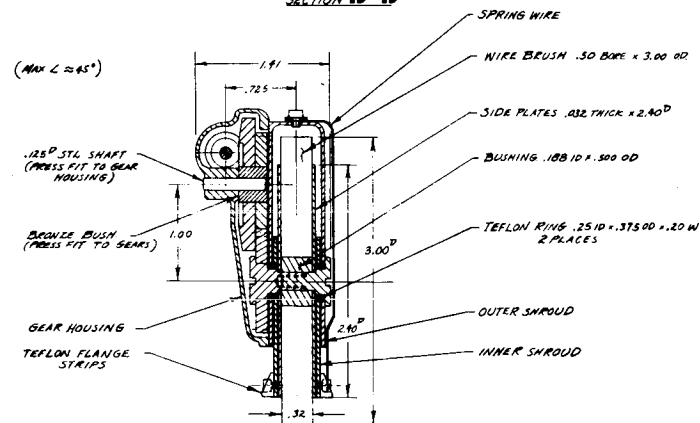


C-8-1

REVISIONS			
SYN	DESCRIPTION	DATE	APPROVED



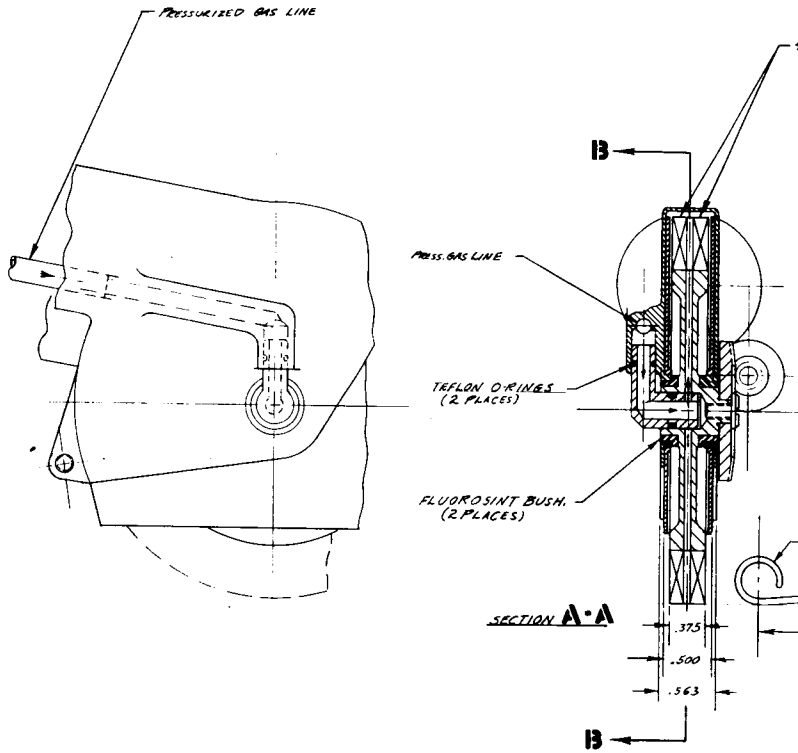
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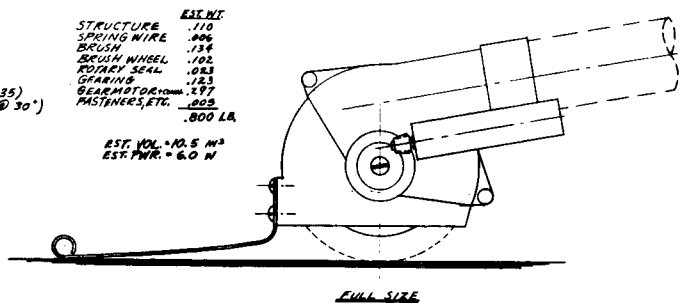
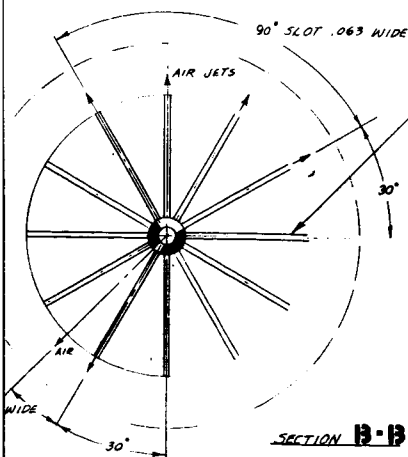
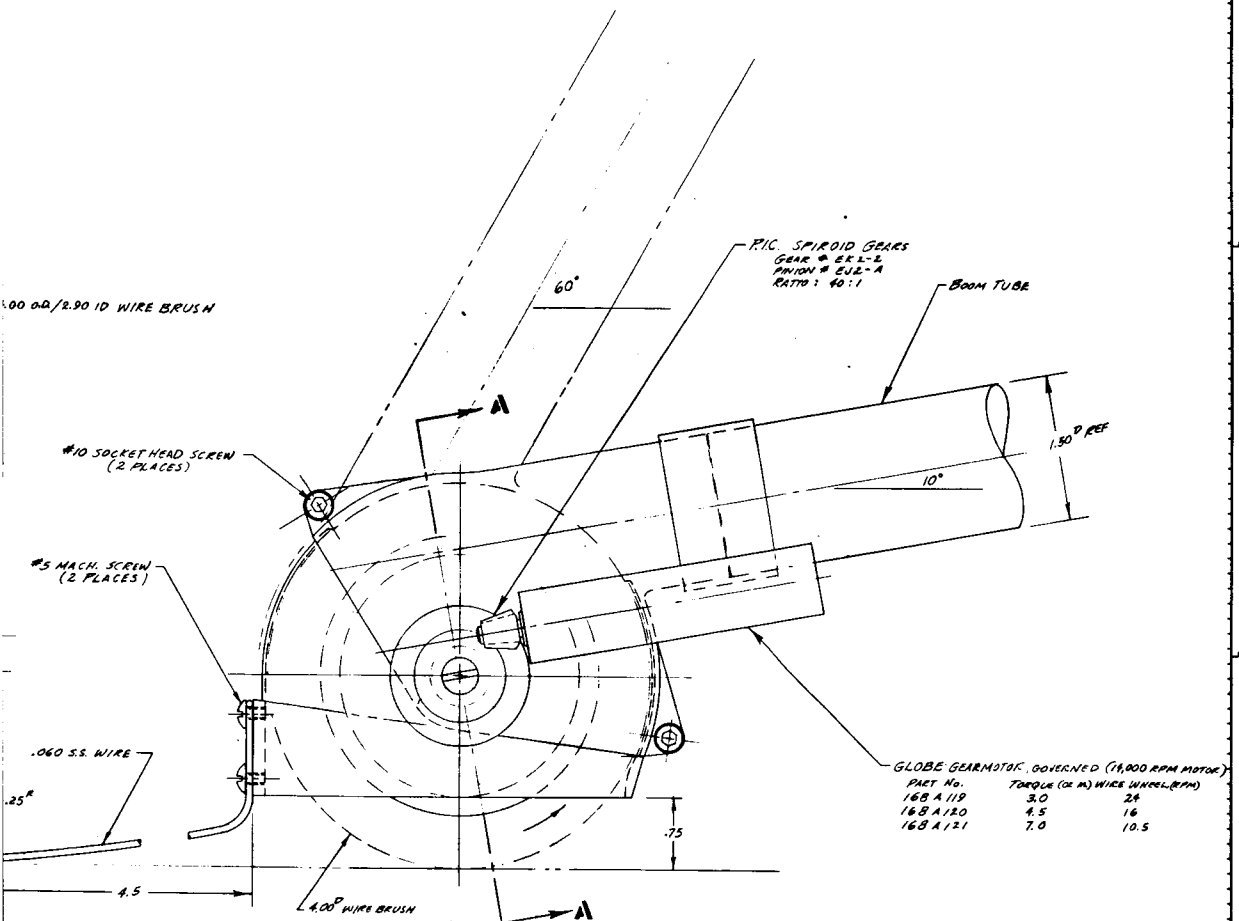
SECTION A-A

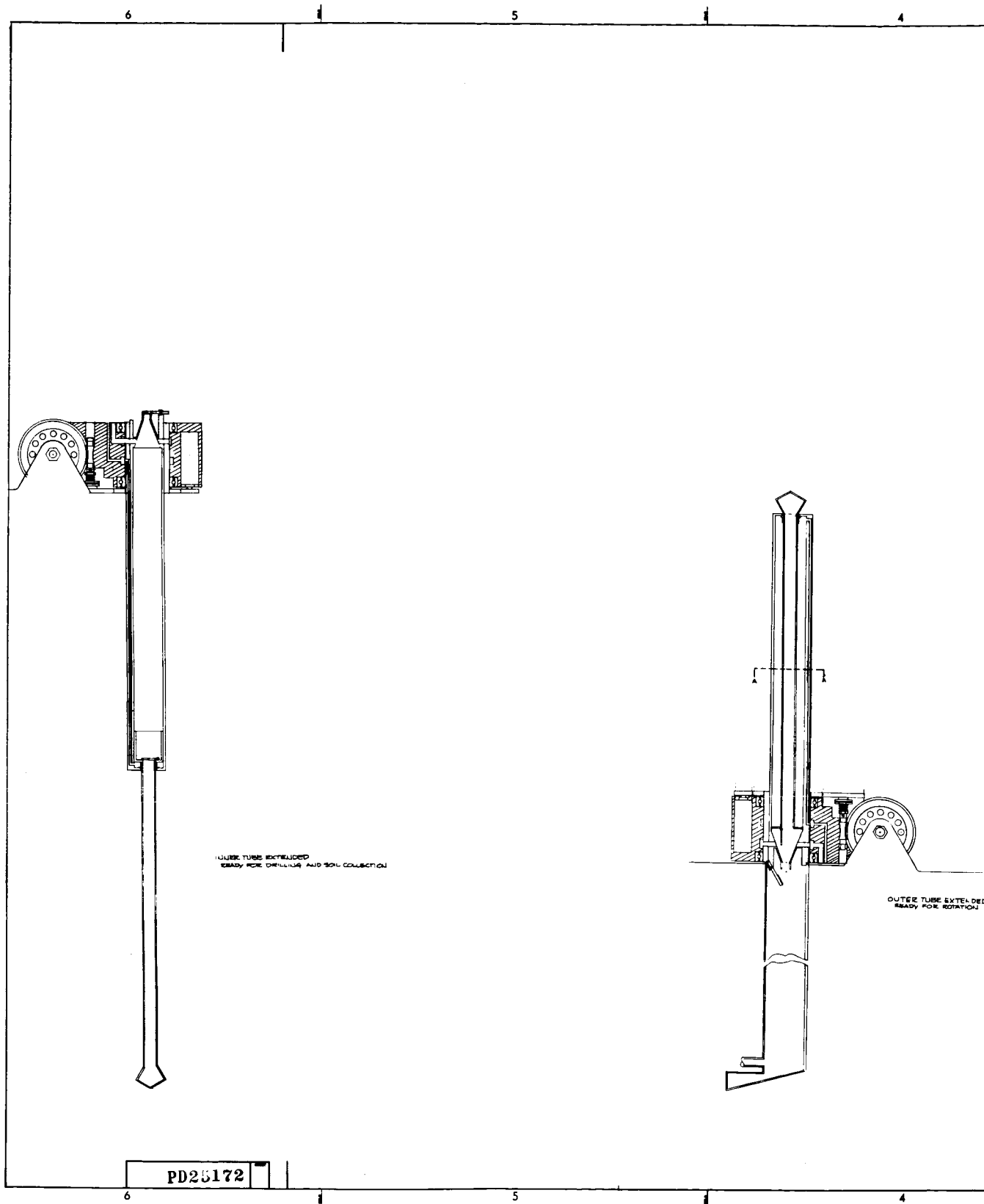
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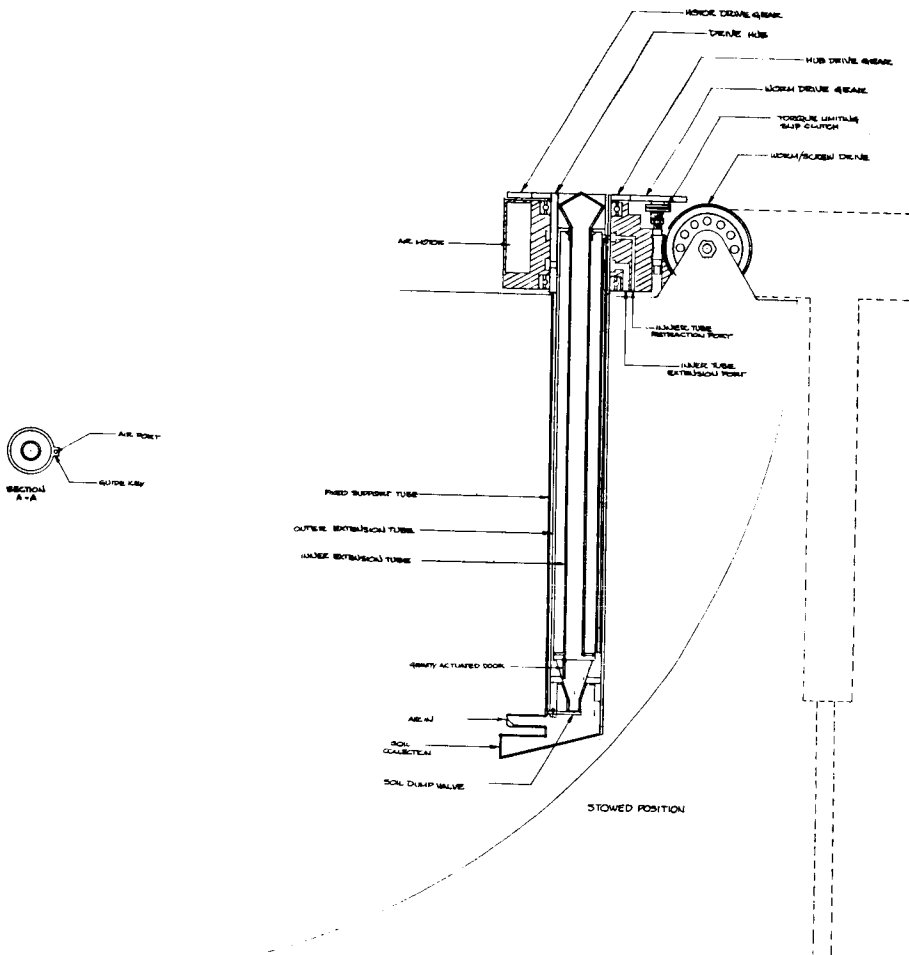


C-9-1

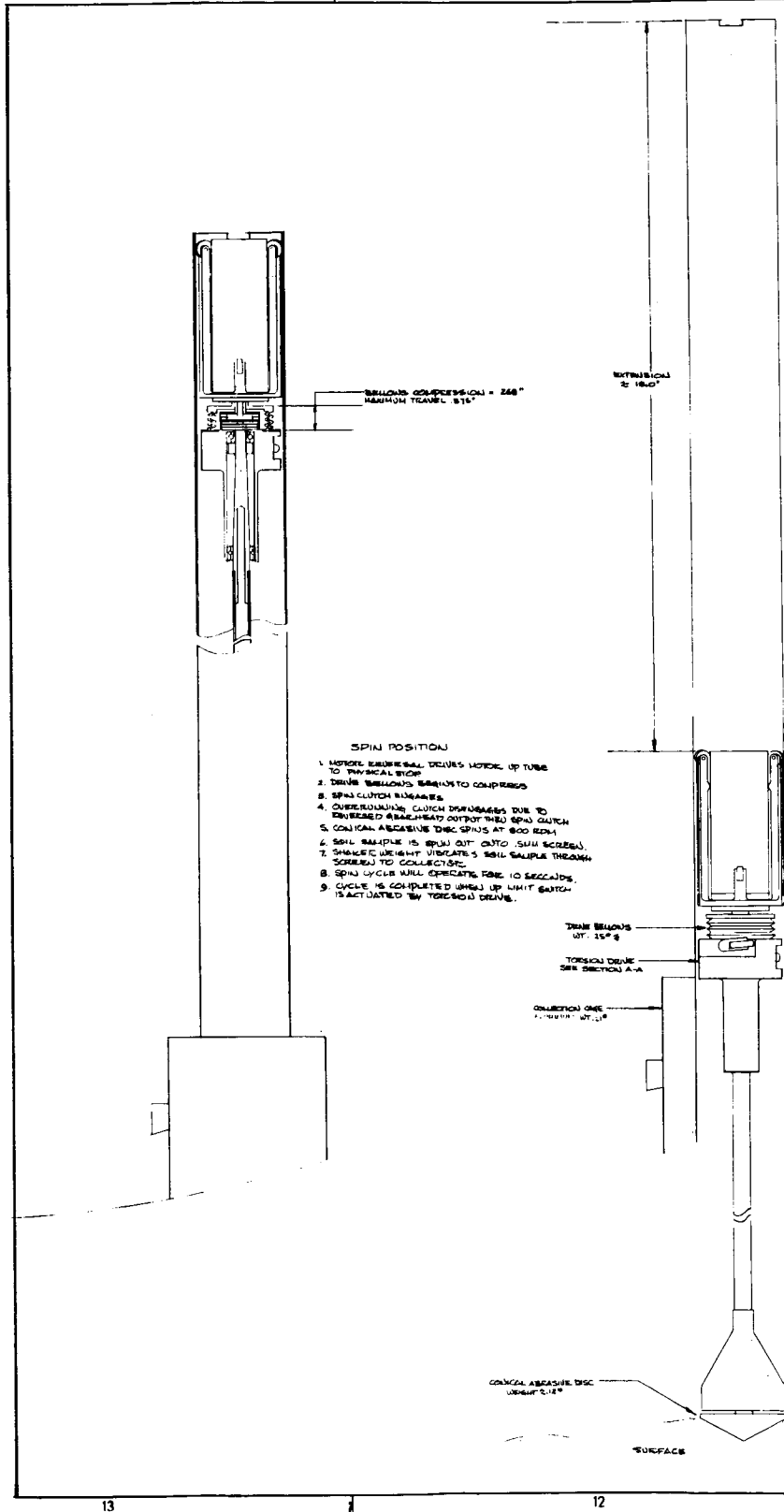
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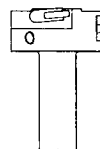
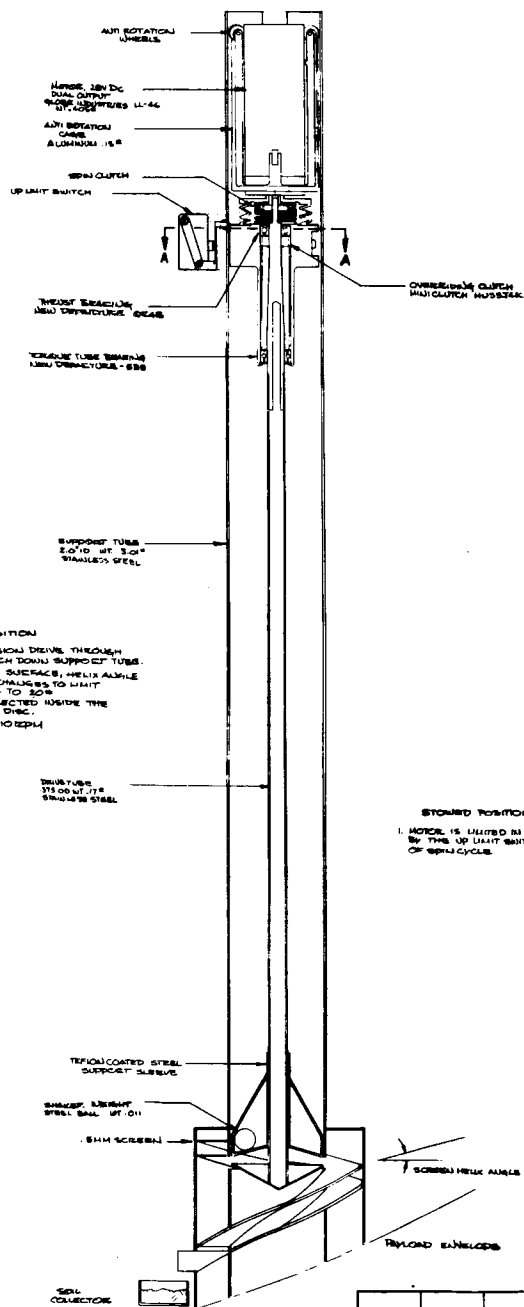
C-10-1



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		TOLERANCES ON DECIMALS	ENGR			
		ANGLES	STRESS			
		DO NOT SCALE THIS DRAWING	WEIGHT			
		RELEASE	PROJ			
		DATE	APPR			
PART NUMBER	NEXT ASSY	USED ON	APPROVED	CODE IDENT NO	SIZE	
APPLICATION	QTY REQD		APPROVED	09205	J	PD25172
				SCALE	WEIGHT	SHEET



C-11-1



TORSION DRIVE
ACTIVE LENGTH 1375"
WEIGHT .168"

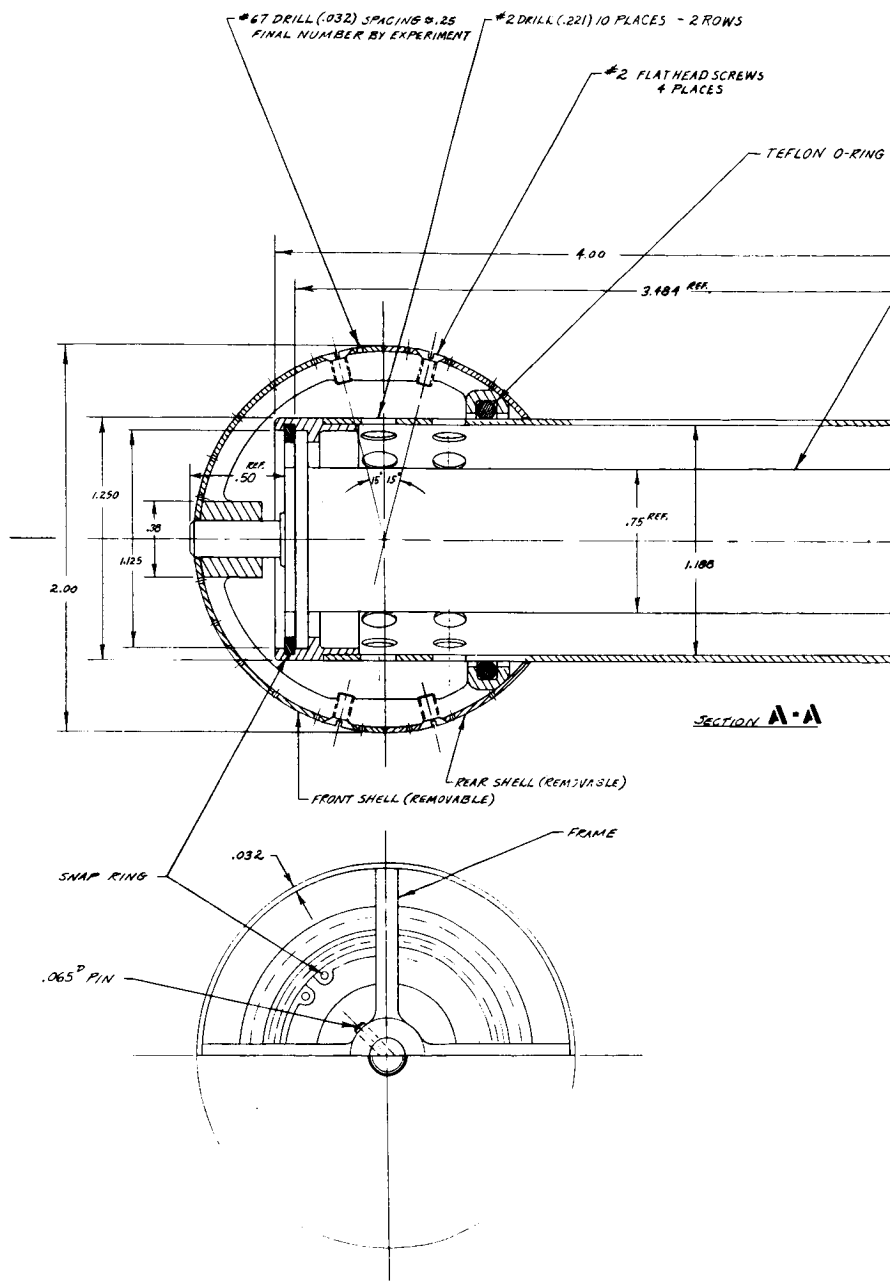
NOTES:

WEIGHT

TORSION DRIVE	.168
SUPPORT TUBE	2.01
DRIVE TUBES	.17
SHANKS WEIGHT	.01
AIR ROTATION DRIVE	.15
DRIVE MOTOR	.40
COILICAL ABRASIVE DISC	.12
COLLECTION CASE	.21
LIMIT SWITCH	.04
DRIVE BELLOWS	.25 B
MANICLUTCH	.12 B
SPIN CLUTCH	.12 B
TOTAL	4.898"

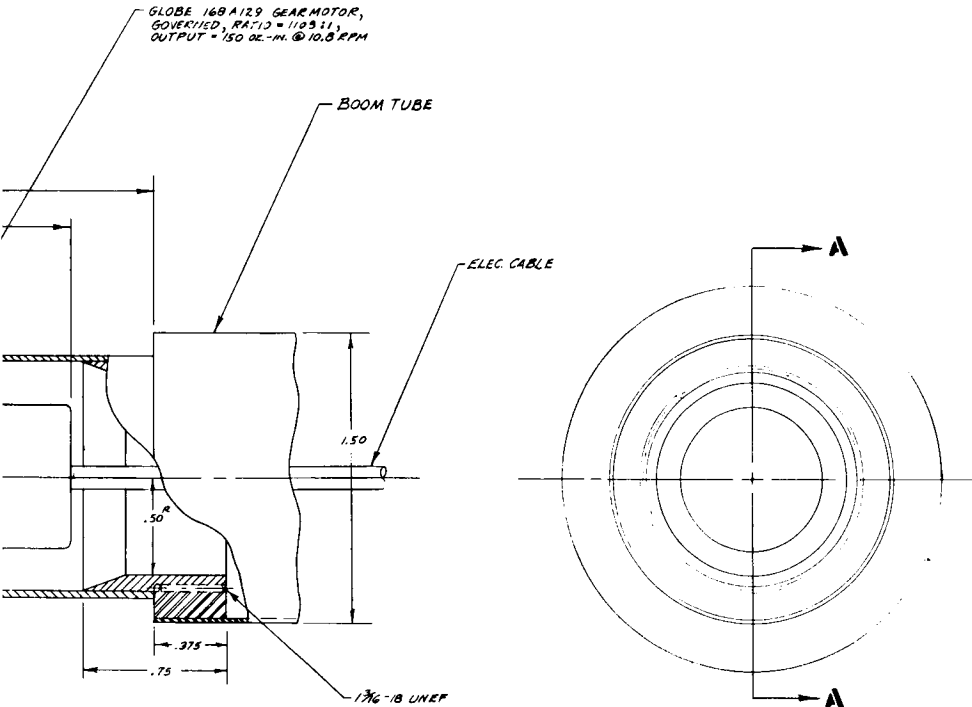
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UNLESS OTHERWISE SPECIFIED						
DIMENSIONS ARE IN INCHES						
TOLERANCES ON						
DECIMALS						
ANGLES						
STRESS						
WEIGHT						
PROJ						
APPR						
SURFACE ROUGHNESS PER MIL-STD-10						
DO NOT SCALE THIS DRAWING						
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NEXT ASSY						
USED ON						
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CODE IDENT NO.						
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PD25166						
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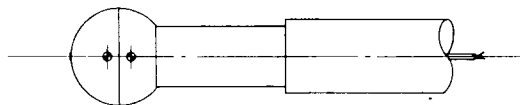
C-12-1

REVISIONS			
REV	DESCRIPTION	DATE	APPROVED



STRUCTURE EST. WT.
 SPHERICAL SHELL .111
 SHELL FRAME .120
 GEAR MOTOR .348
 .6 LB.

EST. VOL. = 7.3 IN³
 EST. PWR. = 6.0 IN



FULL SIZE

QTY REQD	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	SIZE, DESCRIPTION & SPECIFICATION	MATERIAL	ITEM NO.
UNLESS OTHERWISE SPECIFIED			LIST OF MATERIAL OR PARTS LIST		
DIMENSIONS ARE IN INCHES			DRAWN <i>WOODCOCK HANKE</i>		
TOLERANCES ON DECIMALS			CHECK		
1 ± .06			ENGR		
2 ± .03			STRESS		
3 ± .02			WEIGHT		
4 ± .01			PROJ		
SURFACE FINISH			APPR		
PER MIL STD 10			APPROVED		
DO NOT SCALE THIS DRAWING			APPROVED		
RELEASE DATE			CODE IDENT NO. 09205		
QTY REQD			SIZE E		
APPLICATION			SCALE # X		
NEXT ASSY			WEIGHT		
USED ON			SHEET		
NEXT ASSY			PD25173		
FINAL ASSY					

OPERATING SEQUENCE

1. BOOM DEPLOYED VERTICALLY.
2. BOOM EXTENDED BY METELED GAS.
3. BOOM LOWERED TO POSITION.
 - (A) SOLENOID POSITIONING SKIFFLES NORMAL TO BOOM.
 - (B) BOOM POSITION MOTOR OPERATES THROUGH ENERGIZED ELEVATION CLUTCH TO LOWER BOOM.
 - (C) BOOM POSITION MOTOR SHUT OFF BY BOOM LOAD SWITCH OR BY CLUTCH LOCK IN POSITION.
4. SAMPLER OPERATED.
 - (A) POWER CAUSES HEAD TO GO DOWN ON DILUENCE.
 - (B) LOOSENED PARTICLES ARE COLLECTED UP THROUGH SPIRAL SLOTS IN FACE OF HEAD.
 - (C) COLLECTED PARTICLES ARE CENTRIFUGED TO RIM OF HEAD.
5. SAMPLE RETRIEVED.
 - (A) BOOM IS ELEVATED TO VERTICAL (SAMPLER STILL RUNNING).
 - (B) SAMPLER SOLENOID IS SHUT OFF, ALLOWING SAMPLER TO POSITION VERTICALLY.
 - (C) SAMPLER MOTOR IS SHUT OFF, PERMITTING SAMPLE TO FALL DOWN THROUGH HOLLOW DRIVE SHAFT THROUGH HOUSING MOTOR & SOLENOID SHAFTS, DOWN BOOM AND INTO CAPSULE.
 - (D) LOW LEVEL AC POWER TO SAMPLER SOLENOID WOULD CAUSE VIBRATION TO ASSIST FLOW OF SAMPLE.
6. BOOM REPOSITIONED.
 - (A) BOOM POSITIONING MOTOR OPERATES THROUGH ENERGIZED AZIMUTH CLUTCH TO TURN BOOM PRESCRIBED ANGLE IN AZIMUTH (CLUTCH LOCKS IN POSITION).
7. NEW SAMPLE OBTAINED.
 - (A) REPEAT STEPS 2 THRU 5.

A SIMPLIFIED SEQUENCE WOULD REQUIRE RETURN TO FULL VERTICAL FOR SAMPLE RETRIEVAL ONLY AT END OF LAST SAMPLE OPERATION.

SUPPORT REQUIREMENTS

BOOM EXTENSION (GAS OPERATED)

1. 15 FT. @ 1 PSIG.

BOOM POSITIONING (ELECTRICAL)

1. LOWERING AND AZIMUTH: NEGLIGIBLE
2. RAISING FROM HORIZONTAL: 0.7 W/M - 4 WATT MIN.

SAMPLE ACQUISITION (ELECTRICAL)

1. MAXIMUM RATE (WITH 2 INCH DIA. HEAD OPERATING WITH 1 LB. NORMAL LOAD IN SOFT HEAVY MATERIAL): 3.5 WATTS

SAMPLER CHARACTERISTICS

1. HEAD DIAMETER: 2 INCHES (OPTIONAL)
2. RPM: 300 MAX 150 UNDER LOAD
3. SAMPLE CAPACITY (IN HEAD): 10-15 GRAMS.

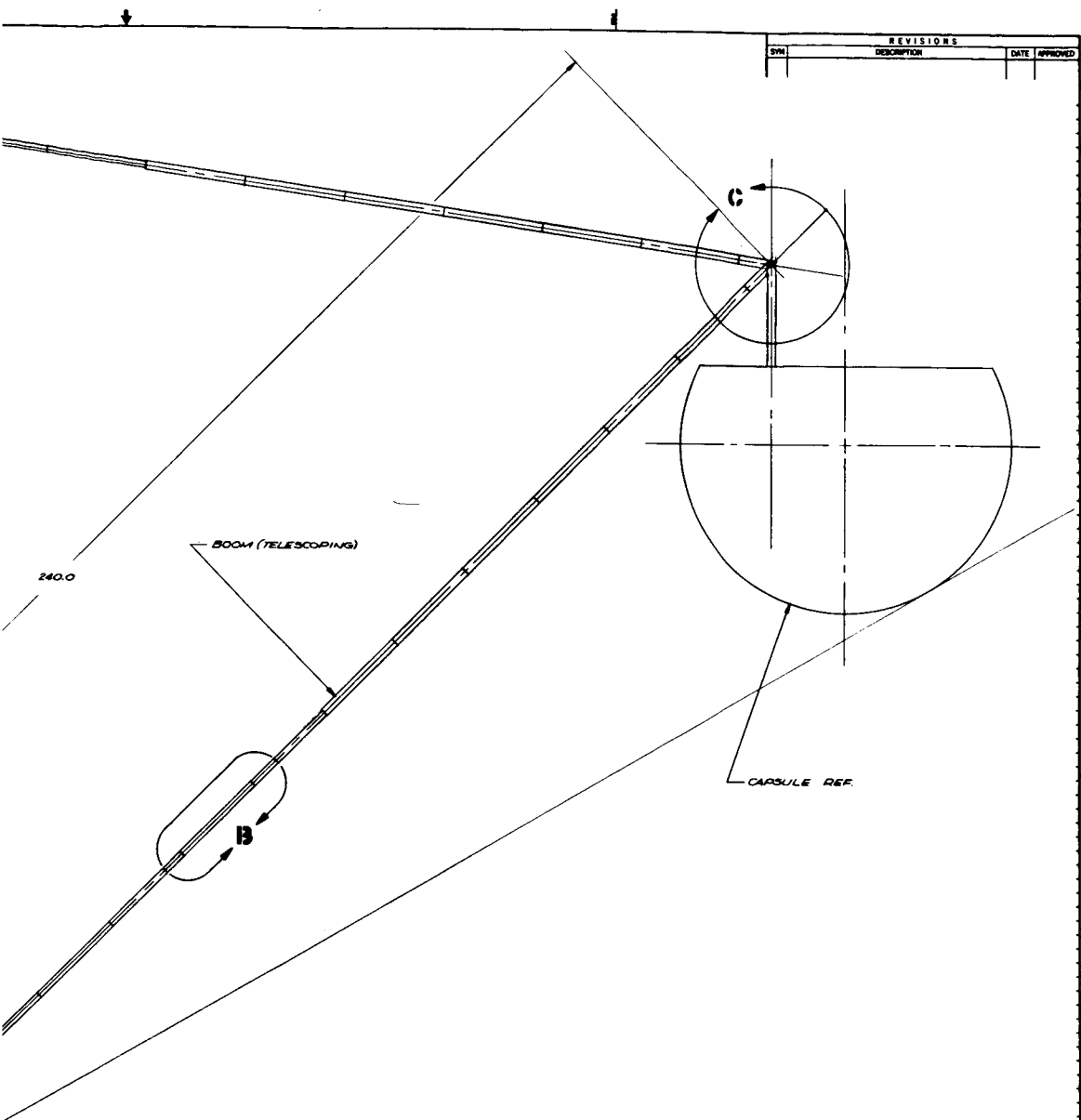
BOOM CHARACTERISTICS

1. LENGTH: 20 FT.
2. DEFLECTION IN BENDING UNDER 1 LB. LOAD AT END: 2 FT.
3. DEFLECTION IN TORSION UNDER 1 LB. FT. LOAD: 1.5 DEGREE

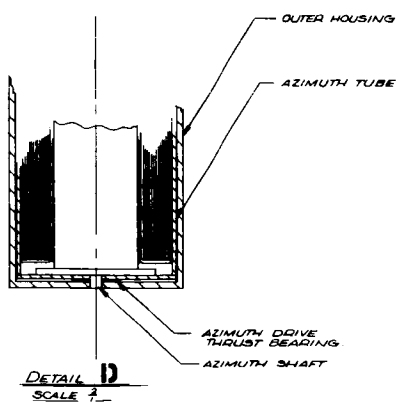
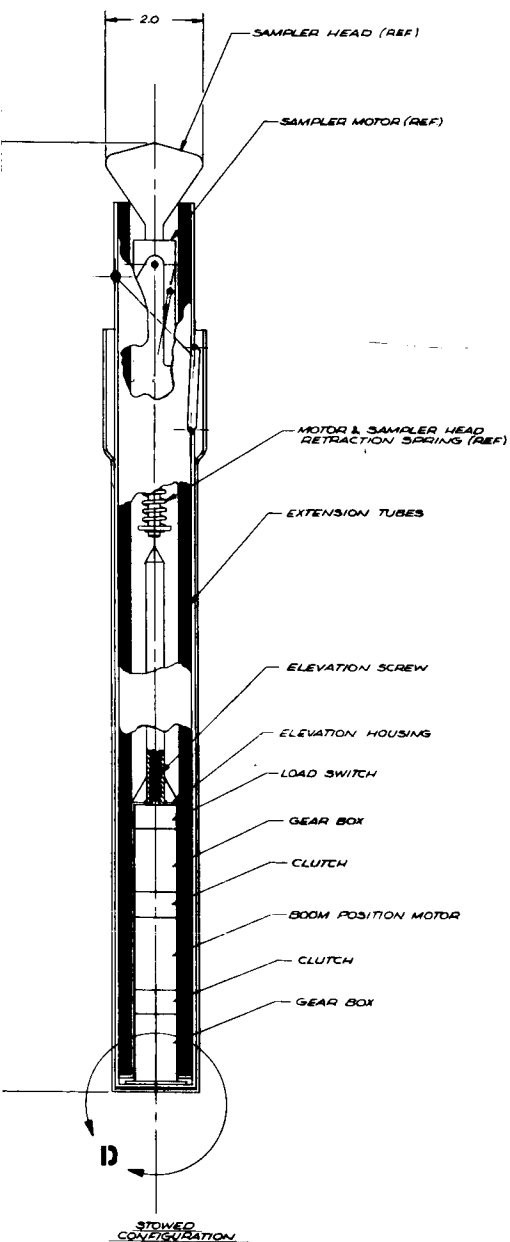
WEIGHTS

1. SAMPLER: 6.55 LB.
2. BOOM: 1.84 LB.
3. BOOM & SAMPLER POSITION GEARMOTOR, CLUTCHES, SOLENOID: 1.75 LB.
4. BOOM HOUSING AND DRIVE LINKAGE: 2.01 LB.
5. TOTAL WT: 6.25 LB.

MOTOR & SAMPLER
HEAD ASSEMBLY

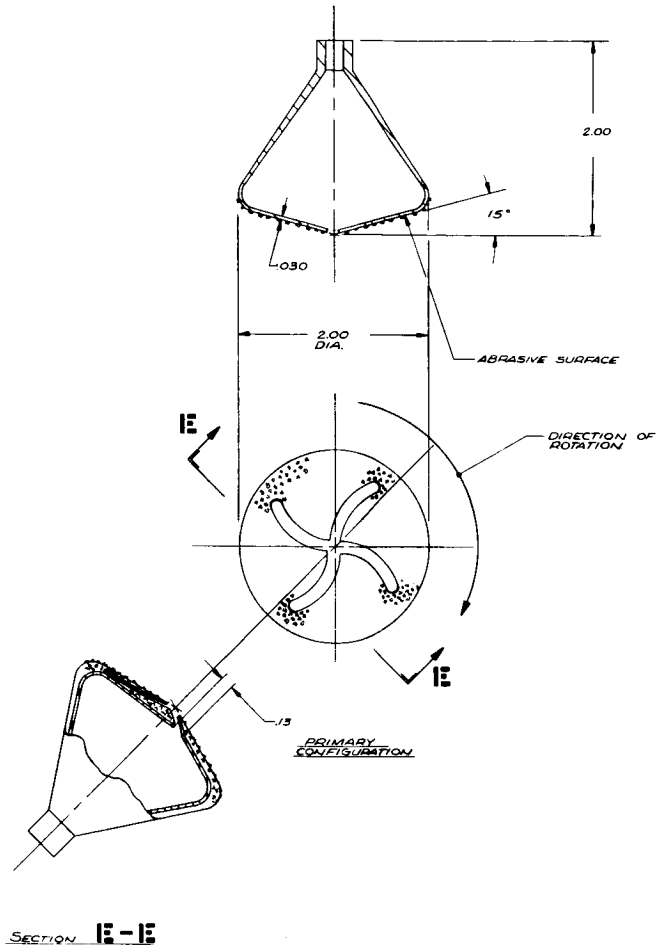


QTY NUMBER		DESCRIPTION NUMBER		REFERENCE NUMBER		NOMENCLATURE		DRAWING NO.		PARTIAL NO.		WORK CENTER NO.	
LIST OF MATERIALS ON PARTS LIST													
UNLESS OTHERWISE SPECIFIED DIMENSIONED PER MIL-STD-883C DIMENSIONS ARE IN INCHES TOLERANCES ON DECIMALS ANGLES .1 .005 .01 .001 .001 .01				DRAWING CHECK CAS74LLO 5-346				FIELD COMPOSITION 100% 100% 100%					
SURFACE TEXTURE DO NOT SCALE THIS DRAWING				LAYER STRIKE WEIGHT PROJ APPD				ABRASIVE SAMPLER HEAD TYPE BD & TELESCOPING BOOM					
PART NUMBER		NEXT ASSY		USED ON		NEXT PART QTY REQD		CODE IDENT NO. 09205		BSR E		PD25177	
APPLICATION		DATE		APPROVED		SCALE		WEIGHT		SHEET / OF 5		1/1	

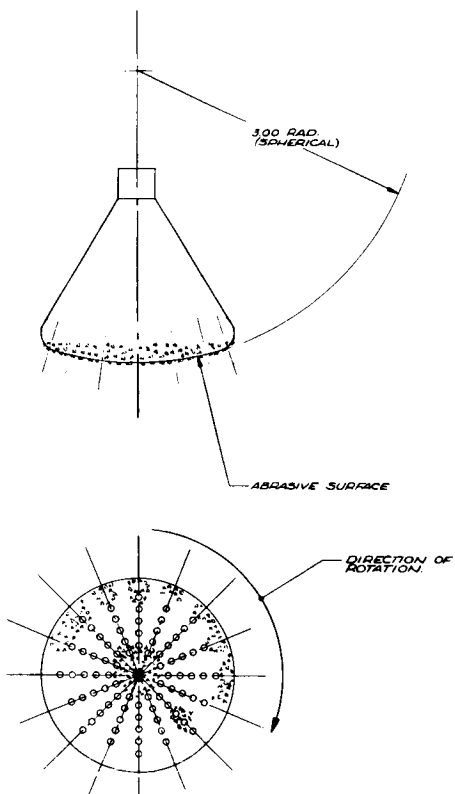


CODE IDENT NO.	SIZE	PD25177
09205	E	
SCALE	WEIGHT	SHEET 4 OF 5

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C-15-1



ALTERNATE
CONFIGURATION

CODE IDENT NO	SIZE	PD 25177
09205	E	
SCALE 1/2"	WEIGHT	SHEET 3 OF 5

PD 25177